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MISSISSIPPI RIVER COMMISSION.

REPORTS ON EFFICIENCY TESTS

—OF—

HYDRAULIC DREDGES

PRESENTED AT THE 89TH AND 90TH SESSIONS OF THE MISSISSIPPI RIVER COMMISSION AND ORDERED PRINTED.

[Being pages 136 to 167 of annual report of Mississippi River Commission for 1903; with addition of records of later tests.]

CONTENTS:

EFFICIENCY TESTS OF HYDRAULIC DREDGES.

OFFICE OF THE SECRETARY MISSISSIPPI RIVER COMMISSION,

ST. LOUIS, MO., *January 23, 1904.*

These tests were undertaken in compliance with the following resolution of the Mississippi River Commission of November 13, 1899, and conducted in accordance with directions from Committee on Dredges and Dredging, dated July 24, 1902:

"That the Secretary be authorized to make such experiments as may seem to be advisable at a cost not exceeding $10,000.00 with the object of increasing the efficiency of the dredges operated by the Commission. These experiments not to be made until after a project and an approximate estimate of cost have been submitted to the Committee on Dredges and approved by the said Committee".

WM. B. LADUE,
Captain, Corps of Engineers, U. S. A.,
Secretary Mississippi River Commission.

REPORT OF ASST. ENGINEER F. B. MALTBY ON EFFICIENCY TESTS OF HYDRAULIC DREDGES.

During the dredging season of 1902 the following-described tests of the efficiency of the engines, boilers and sand pumps on the various dredges belonging to the Mississippi River Commission were made. The tests were made in obedience to the following instructions, prepared by the committee on dredges and dredging of the Commission:

1. Such tests shall be made as may be necessary to determine the efficiency of boilers, engines, and sand pump of each of the dredges. The relative efficiency of the several types of jet pumps, with due consideration of the results required in economical dredging work, should also be carefully determined.

2. As a basis for determining the mechanical efficiency of engines and pumps under working load, it is necessary to first determine their frictional horsepower when running at normal speed without load.

3. The pump tests shall be made by pumping water with the intake submerged

to the normal depth and the pump running at normal speed, and also at known speeds both higher and lower than the normal to ascertain the effect of variations in speed.

4. In order to ascertain as far as practicable the effect of the form of suction head, tests shall be made both with and without the suction head, where these are so attached as to be readily removable.

5. Each test shall embrace the determination of the indicated horsepower of the engines, the number of revolutions of pump per minute, the velocity of flow in suction and discharge pipe, the suction and discharge pressures.

6. In addition to the pressure gauges now in use, mercury manometers should be attached to suction and discharge pipes near the pump for the accurate determination of suction and discharge pressures.

7. The velocity of flow in suction and discharge pipes shall be carefully measured and their determinations should be made at several points in the cross section of the discharge pipe, so as to determine whether or not the whole of the discharge section is effective under normal pumping conditions. This test can, however, only be made when pumping sand; but it would not interfere materially with the regular field work if done when the dredges are in operation. Pitot tubes are recommended for use in making velocity observations.

8. The loss of head due to friction in discharge pipe shall be determined. It is also desirable to carry this investigation further, if found practicable, so as to include the effect of curved sections, rough joints, etc.

9. It is desirable to measure, as far as practicable, the relative efficiency of the double and single intake to ascertain whether the flow of two columns of water from opposite directions and meeting at the center of the pump tends to materially reduce the efficiency.

10. In conducting the above required investigations other lines of inquiry will doubtless be suggested, and if they promise results of value they should be followed up.

11. When the required observations have been completed, they shall be carefully studied and compared with a view to determine the most efficient type of engine and pump now in use and how the best of these could be improved upon in future construction.

12. The results of the above investigations shall be embodied in a report giving in detail the type and form of boilers, engines and pumps examined and the observations made in each case, with a summary showing from the results which type or combination of types is the most efficient and best for the conditions met with in the Mississippi River.

13. It is intended that the investigations and experiments called for above will be made at such times as the dredges are not otherwise employed, as when lying at the bank waiting for suitable stage of water or at the close of the coming ing dredging season before being laid up for the winter. It is therefore desirable to have such preparations made in the way of instruments and measuring appliances and attachments as may be deemed necessary before going into the field.

GENERAL DESCRIPTION OF TESTS.

It was thought desirable in making these tests to obtain actual working results and to ascertain what was actually being done, rather than what it might be possible to do. With this idea in mind the tests were all made with the ordinary regular

crews working in the ordinary manner. No alterations or preparations were made on the boilers or machinery except those necessary for attaching the measuring appliances.

All the plant was in good ordinary working condition, as none of it had been used to any great extent since the ordinary overhauling made during the lay-up season. The tests were made at such times as the dredges were not otherwise engaged and at such times as could be spared by the superintendent of dredging operations from his other duties.

In general the same observers were used throughout all the tests as far as practicable; all weighing scales and gauges used were carefully tested; thermometers, pyrometers, and calorimeters were of a high grade, new, and were not tested: indicator springs were calibrated.

The information and results desired, as indicated in the above instructions, may be divided under the following distinct heads and will be treated separately:

1. The efficiency and comparison of the boilers.
2. The efficiency and comparison of the main pumping engines.
3. The efficiency, capacity, and comparison of the sand pumps.
4. The effect of the form of the suction heads.
5. The efficiency and comparison of the jet pumps.
6. The loss of head in the discharge pipes due to friction and other causes.
7. The determination of the effective cross section of the discharge pipe when pumping sand.

BOILER TESTS.

In making tests of the boilers no especial preparations were made for the tests. The boilers and settings were all in good condition. The regular firemen on each dredge were used and watches changed in the usual manner. The only instructions given the firemen were to fire as regularly as possible and not to shut the draft doors, as the engineers would take care of any excess of steam generated. In nearly all cases the pumping engines and auxiliary engines were run in the ordinary manner in order that the demands on the boilers might be the same as in ordinary service. It is believed that the efficiencies obtained show fairly well the actual results obtained in regular operation.

METHODS.

The method of starting and stopping a test was as follows: The plant was operated till everything was running smoothly and regularly. The deck of the fire-room was cleaned of coal, ash pits cleaned, fire examined by the superintendent, the height of the water in the gauge glasses noted, and the word given to start the test. During the last fifteen minutes of the test the fires were closely watched by the superintendent and brought into as near the same condition as when starting the test as possible.

COAL.

The coal was what is known as "Pittsburg lump" and was taken directly from barges to the scales. No tests were made of the coal. It was all, however, practically "air dry." It was all carefully weighed on tested scales and delivered to the firemen in lots sufficient to last approximately one hour. All ashes were removed and weighed dry.

FEED WATER.

Water was pumped from the river into a cylindrical measuring tank about 3 feet in diameter and 4 feet high. It was emptied through a 6-inch gate valve near the bottom. The vertical height drawn out each time was measured by means of a large wooden float, varnished to prevent absorption of water, bearing a staff graduated to hundredths of a foot. This staff passed through a hole in a board across the top of the tank. The tank was first filled, water supply shut off, and the position of the staff read; the water was then drawn off, care being taken that the surface of the water did not go below the top of the opening leading to the discharge valve, the discharge valve closed, and the staff again read. The weight of the water per foot, as indicated by the staff readings, was determined by drawing water from the tank at various heights, observing the amount so drawn off on the staff, and weighing it on standard scales. The mean of twenty observations was used in computing the weight per foot. The measuring tank discharged into a suction tank supplying the boiler feed pump. The temperature of the feed water was measured with a high-grade thermometer at a point in the feed pipe just before entering the boilers. The height of the water in the boilers was kept constant as

nearly as possible and pains taken to have it exactly the same at the beginning and end of the test. All blow-off valves and pipes and pipes from the feed pump were examined and made tight.

CALORIMETER.

The quality of the steam was determined by means of a Carpenter throttling calorimeter attached to the main steam pipe as close to the boiler as possible.

FLUE GASES.

The temperature of the flue gases was determined by two horizontal graphite pyrometers furnished by A. S. Aloe Company. They were not calibrated. They were inserted in the stacks just above the breeching.

The temperature of the feed water and readings of the calorimeter and pyrometers were taken every half hour, and the mean readings used in computing results.

Grate and heating surfaces were obtained from careful measurements of each boiler.

A table showing data and results obtained is appended hereto.

DISCUSSION OF RESULTS.

It is believed that the results on the *Gamma*, *Delta*, *Epsilon*, *Zeta*, and *Kappa* show good average working conditions and are fair examples of the efficiencies of the boilers of their class.

On the *Iota* considerable trouble was experienced in operating the engines, and for this reason it was not possible to consume the steam regularly. On the *Flad* the same difficulty was experienced, though to a less extent. The comparatively low efficiency of the latter is believed to be due to the firing, which was by far the poorest during any of the tests. The mean efficiency of the Mississippi River boilers per pound of combustible is 9.10 pounds of water evaporated, as against 9.34 for the water-tube boiler on the *Delta*. The high efficiency of the boilers on the *Gamma* is due, it is thought, to the relative large ratio of heating surface to grate surface and the high rate of combustion per square foot of grate surface. The 5-flue boilers do not seem to be more efficient than the 4-flue, and, in fact, as between the *Epsilon* and *Zeta* and the *Iota* and *Kappa* the advantage seems to lie the other way. For ease of cleaning and danger of burning the points of superiority all lie with the 4-flue boilers. The tests show that the Mississippi River type of boiler, while possessing the advantages of simplicity, ease in cleaning and repairing, is also an economical boiler in operation.

These tests also show incidentally the importance of a feed-water heater. The *Gamma* is not supplied with a heater, water from the river being pumped directly into the boilers. It will be noted that the correction applied to the quantity of water evaporated into dry steam to bring the feed water to 212 degrees is nearly 19 per cent of the total. A feed-water heater using the exhaust, which now goes to waste from the auxiliary engines, as on the *Della*, *Iota*, *Kappa*, and *Flad*, assuming that the same temperature of feed water would be secured, would reduce this to a little over 9 per cent; or, in other words, the same amount of coal would evaporate nearly 10 per cent more water.

Sketches showing the characteristic features of the boilers and settings are appended.

EFFICIENCY OF MAIN PUMPING ENGINES.

It was not thought necessary to determine the efficiency of the engines on all the dredges, as many of them have the same general characteristics. All are engines of standard manufacture, modern, high speed, compound, and all use about the same steam pressures, the most important difference being that the *Gamma*, *Delta*, *Iota*, *Kappa*, and *Flad* are equipped with condensing engines, while the engines on the *Epsilon* and *Zeta* are noncondensing. To determine the relative efficiency of these two types, tests were made to determine the amount of water used per indicated horsepower hour by the condensing engines on the *Kappa* and the noncondensing engines on the *Epsilon*.

METHODS.

It was arranged that all the steam generated by the boilers should be used by the engines under test alone on the *Epsilon* and by the engines and condenser on the *Kappa*. Steam for operating the feed pumps and auxiliaries was furnished

from an outside source. Especial care was taken to prevent leaks or to collect the drips from any that existed. The feed water was measured in the same manner as described under tests of boilers. The quality of steam was determined by the same calorimeter, attached to the steam pipe just above the cylinders. Indicators were attached to all four cylinders and cards taken simultaneously. Indicator cards and calorimeter readings were taken every half hour and the means were used in computing results.

DESCRIPTION OF ENGINES.

The engines on the *Epsilon* are "Ideal" tandem compound, a pair on each end of the main shaft of the pump. The cylinders are 16 inches and 26 inches in diameter by 18 inches stroke; piston valves on high-pressure cylinders and balanced slide valves on low-pressure cylinders. There is no receiver between the cylinders, and the cylinders have no steam jacket, but are well lagged with mineral wool. They have automatic fly-wheel governors. The normal speed is from 172 to 176 revolutions per minute, and they developed a mean indicated power of 731.82 horsepower and consumed 26.06 pounds of water per indicated horsepower hour. They were made by A. L. Ide & Sons, of Springfield, Ill.

The engines on the *Kappa* were built by the Bucyrus Company, of South Milwaukee, Wis. They are also tandem compound, direct connected to the pump shaft. The cylinders are 15½ inches and 30 inches in diameter, 24 inches stroke. High-pressure valves are of the piston type, and the low pressure balanced slide valves. The cylinders have no steam jackets, but are well lagged. They are supplied with automatic fly-wheel governors. The normal speed is about 130 revolutions per minute, and they develop a mean total horsepower of 852.01 and consume 21.42 pounds of water per indicated horsepower hour. This steam consumption includes the amount used by the condenser on the *Kappa*.

It is thought that the difference of 4.64 pounds of steam per hour per indicated horsepower in favor of the engines on the *Kappa* represents in a fair way the economy of the condensing engines over the noncondensing of this size and type. The boiler tests of the *Epsilon* show an evaporation of 7.96 pounds of water per pound of coal, or to evaporate the excess of 4.64 pounds of water per indicated horsepower hour on the *Epsilon* requires, when developing 730 horsepower, 551 pounds of coal per hour, or about 13 per cent of the total amount consumed by the dredge. This, at present prices, costs nearly $19 per twenty-four hour day for coal only, not including the cost of handling it.

A table showing in detail the data and results of this test is appended.

THE CAPACITY AND EFFICIENCY OF THE SAND PUMP.

In the design of a hydraulic dredge one of the principal features to be considered is the design of the sand pump, its efficiency and capacity, and it is evident from the resolution of the committee that the determination of the efficiencies and capacity of the various forms of pumps in use was considered the most important of the tests to be made. On the seven dredges under consideration all the pumps are of different design except on the *Kappa* and *Flad*, which are identical. The *Epsilon* and *Zeta* were originally of the same design, but the pump on the *Epsilon* has been supplied with a closed or shrouded runner, and on both the *Epsilon* and *Zeta* the number of blades has been reduced from 7 to 5. The details of each pump will be referred to later in the discussion of results.

METHODS—THE PITOT TUBE.

In determing the capacity of the pumps one of the first points to be decided upon was the method to be employed in determining the velocity of flow through the discharge or suction pipes. The instructions recommended the use of Pitot tubes. After a careful study of the subject the writer believes that there is no other simple, inexpensive, and practical method of measuring the high velocities in the 32 and 34 inch discharge pipes in use with the dredges. The publication in the Transactions of the American Society of Civil Engineers for April, 1902, of a paper on experiments on the flow of water in pipes, and especially the discussion of the original paper, was most timely. A large portion of the original paper and its discussion by members of the society is devoted to the Pitot tube, and attention is invited to it for a complete discussion of its merit as an accurate instrument for measuring the velocity of flowing water.

This accuracy seemed to be fully conceded, with tubes properly constructed, and it remained to design a tube that could be inserted readily into the suction or

discharge pipes that would be strong enough to stand the rough usage it would meet and still show accurate results. All the tubes were made in the machine shops connected with the dredges, as well as the gauges with which they were connected. Nine tubes in all were constructed, but the inherent defects of the design of all but Nos. 1, 3, 8, and 9 were soon apparent and they were discarded.

DESCRIPTION OF TUBES.

A sketch of tubes Nos. 1, 3, and 8 is appended hereto.

Tube No. 1 consists of two pieces of brass tubing, one-fourth inch inside diameter, inclosed in a piece of pipe $32\frac{1}{2}$ inches long and turned to $1\frac{5}{16}$ inches outside diameter. One of the small brass tubes is bent at its lower end, and below the end of the outside inclosing pipe, to a right angle and the plane of the opening made truly parallel with the upright pipe. This forms the impact opening or point. The other brass tube is brazed onto a solid brass piece, circular in section, and placed at right angles to the tube, the upstream end is turned down to a sharp point and its end is even with the opening in the impact point and below it; this brass point has a one-sixteenth inch hole drilled on each side, connecting with the interior of the upright brass tube, and forms the static or pressure point. The upper ends of the brass tubes have one-fourth inch air cocks, which are connected by pieces of one-fourth inch rubber tubing, about 4 feet long, to the gauges.

The small brass tubes are held in the outside tube by lead or cement poured around them. The outside tube slides through a stuffing box, which is screwed into a hole tapped in the side of the suction or discharge pipe with a $1\frac{1}{2}$-inch pipe tap. The top of the tube has a handle set carefully at right angles with the plane of the impact opening. This arrangement permits the point of the tube to be placed at any point along the diameter of the pipe in which velocity is being measured and permits "traversing," or the determination of the velocity curve across the pipe.

Tube No. 3 is very similar to No. 1, except that the impact point is below the static point.

Tube No. 8 has the same sized vertical outside pipe to permit of the use of the same stuffing boxes. At its lower end it is joined at right angles to a 1-inch pipe about 18 inches long, drawn down at the point to the size of a one-fourth inch brass tube, which is brazed into it and forms the impact point. This small brass tube runs inside the large pipe to the top, where it is provided with a one-fourth inch cock. The horizontal pipe has three one-sixteenth inch holes drilled in its side for determining the static pressure, the interior of the pipe being connected through the upright pipe and one handle to a one-fourth inch air cock. Tube No. 9 was made as nearly like No. 8 as could be done by a skilled machinist.

In the design of the tube it was thought that the fact that the impact point, when formed of a surface of revolution, will convert velocity head into static head, in accordance with the law $v=\sqrt{2gh}$, had been thoroughly proven in the discussion by Mr. W. M. White of the paper above referred to and some experiments made by him. (See Journal of the Association of Engineering Societies, August, 1901.) This point being accepted, it only remains to show that the static or pressure point of the tube used gives the true static pressure, or, in other words, that there is no suction action due to the flow of water past the static openings. If this is true, the tube will have a coefficient of unity in the formula $v=c\sqrt{2gh}$, where c is a coefficient and h is the observed difference of pressure shown by the static and impact points of the tube.

The most conclusive method of demonstrating this fact would be to rate the tube under the same conditions under which it was used, but the difficulties of rating a tube in a 32-inch pipe having a velocity of flow of from 14 to 25 feet per second and under a pressure of from 18 to 30 feet of water can readily be realized.

It is known that tubes 1 and 3 have a coefficient less than unity, but it is believed that the coefficient of tubes 8 and 9 is unity. The following observations are presented in support of this belief. The tube was rated in open running water having a velocity of about $3\frac{1}{2}$ feet per second by comparing it with floats. About 80 floats were run and 600 observations were made with tube No. 9. In this case the tube was suspended in the river with the point about 12 inches below the surface and facing the current. The impact point and static point were attached to opposite ends of an inverted U tube of glass, from which the air could be partly exhausted in order to bring the water surface in the tubes high enough to be read.

The floats were 24 inches long, weighted to stand vertical in the water, and three-quarters or more of their length was submerged. The time required to pass over a course 40 feet long was observed, the tube being about one-third the

distance from the upper end of the course. In the following table the velocity given by the floats is in each case the mean of ten observations and that given for the tube is the mean of about seventy-five observations.

	Velocity by float.	Velocity by tube.
No. 1	3.316	3.420
No. 2	3.284	3.279
No. 3	3.517	3.393
No. 4	3.419	3.358
No. 5	3.405	3.502
No. 6	3.212	3.228
No. 7	3.362	3.354
No. 8	3.332	3.218
Mean	3.356	3.343

The means show a coefficient of unity within about one-third of 1 per cent.

If tubes 8 and 9 have coefficients of unity they should both give the same velocity in the same pipe at the same time.

To demonstrate this they were placed in the same discharge pipe, 50 feet apart, and three sets of observations made; the position of the tubes was then reversed and three sets of observations again taken, with results as follows. Each set is the mean of ten readings.

Velocity by tube No. 8.	Velocity by tube No. 9.
25.370	24.500
24.160	25.600
25.770	25.140
[a]25.360	[a]25.190
25.370	24.840
25.550	25.020
Mean, 25.263	25.048

[a] Reversed positions.

The means agree within about eight-tenths of 1 per cent.

That the accuracy of measurements by the tubes is not affected by pressure was determined as follows:

Tube No. 8 was placed in the discharge pipe on the second ponton in the pipe line of the dredge *Epsilon*, where the average static pressure was 17.50 inches of mercury; tube No. 9 was placed in the discharge pipe on ponton No. 9, 350 feet from tube No. 8 and where the average pressure was 3.64 inches of mercury. Simultaneous readings were taken on gauges connected with each tube and at different points across the pipe.

The mean velocity determined by tube No. 8 from 170 observations was 22.321 feet per second. The mean determined by tube No. 9 was 22.351 feet per second, a difference of one-tenth of 1 per cent.

The pressure indicated by the static openings in tube No. 9 was compared with piezometers in the sides of a 32-inch pipe as follows:

Four holes were drilled and tapped for a one-fourth inch pipe on the four "quarters" of the 32-inch discharge pipe on the dredge *Kappa*, all being in the same vertical section. Care was taken to have the axis of these holes normal to the surface of the pipe. Ordinary one-fourth inch air cocks were screwed into the holes, their ends projecting inside the pipe slightly. These were afterwards filed down carefully to present a perfectly smooth surface flush with the inside of the pipe. It was thought that if there was any inaccuracy in the pressures indicated by the piezometers it would not be probable that the errors would be the same on each one, or if the pressures indicated by each piezometer were the same all would be correct. Each of the piezometers was in turn connected with every other one through a differential gauge and in no case could any difference in pressure be observed.

Tube No. 9 was then inserted into the same vertical section as the piezometers and with the point of the tube at the center of the pipe. The static side was in turn connected with each piezometer through a differential gauge and in no case

could any difference of pressure be observed. The pressure given by the four piezometers and the static side of the tube was exactly the same.

This experiment, with some others taken with the point of the tube at different points along the diameter of the pipe, show, incidentally, that the pressure is the same throughout any section across a pipe and that the Bernouilli theory is not true when applied to a section.

It is conceded that the above observations do not demonstrate absolutely that the coefficient of tubes 8 and 9 is unity, but in connection with the discussion of the paper before referred to, it is believed that they furnish sufficient evidence to warrant us in assuming that their coefficient is unity, and the observations made with these tubes have been so used in reducing results.

Tubes Nos. 1 and 3 were compared with tube No. 8 by placing them in the same discharge pipe 50 feet apart. Three sets of observations of ten each were taken and the position of the tubes in the pipe reversed and three sets more taken. The results show the coefficient of tube 1 to be 0.930 and of tube 3 to be 0.8915, and these values have been used in reducing observations made with these tubes.

A brief description of the other tubes experimented with and abandoned may be of interest. A sketch is attached, showing the points only, the inclosing pipe, stuffing box and attachments for connecting with gauges being in each case similar to tubes 1 and 3.

Tube No. 2 was composed of an impact point similar to tubes 1 and 3. The pressure point was a simple vertical brass tube with the lower end open and the plane of the opening parallel with the current. This form of pressure opening showed the greatest amount of suction action of any tube experimented with. In fact, the suction was greater than the impact force on the impact point, as was evidenced by the fact that when the impact point was turned to point downstream it still showed an apparent difference of pressure, due to impact, about half as great as when pointing against the current.

Tube No. 4 was tube No. 2 with the lower end of the static point plugged up and with openings on the side.

Tube No. 5 had impact and pressure points similar, but pointing in opposite directions.

Tube No. 6 was very similar to No. 1, except that larger tubes were used, the impact point being filled with a plug having a hole one-eighth inch in diameter.

The coefficient of these tubes, except No. 6, was not obtained, as at the time they were made we were in search of a tube having a coefficient of unity and it was quite evident that these tubes did not possess that qualification.

In comparing the various tubes with each other they were at first placed in the discharge pipe 4 feet apart. It was apparent that the one lowest downstream was affected by the one above it to a very considerable extent. The distance apart was increased to 8 feet, with still an appreciable disturbance. The distance was then increased to 50 feet, and no disturbance could be detected, and the coefficients of tubes Nos. 1 and 3, as compared with tubes Nos. 8 and 9, were determined at this distance, as has been described.

THE GAUGES.

Sketches show the two different styles of gauges used.

The differential gauge is a smple U-shaped glass tube, one-fourth inch inside diameter, provided with a connection at each end for attaching rubber tubes from the Pitot tubes. The whole was securely fastened to a board, with raised edges to protect the glass from injury. A paper scale, divided into inches and tenths, was fastened to the board behind the tubes, and both the scale and board were varnished over to prevent injury from water. The open gauge was also a simple U-shaped glass tube with one leg longer than the other and with an attachment for connecting with a rubber tube at one end only. The attachments for rubber tubes were made of cast iron. The holes, where they were joined to the glass tubes, were counterbored about three thirty-seconds of an inch larger than the glass and this space filled with Portland cement, which made a most satisfactory joint. The gauges were connected to piezometers or Pitot tubes with cloth-inserted rubber tubing one-fourth inch inside diameter and not over 4 or 5 feet in length. Considerable patience and ingenuity were required to remove all the air from the tubes and the gauges above the mercury. It was found that a small amount of air in the tubes would affect the readings of the gauges quite materially. The glass tubes were not calibrated, but a careful examination revealed no appreciable difference in their size. Mercury was used in the gauges and the specific gravity was assumed to be 13.5. It is believed that the possible errors due to slight differences in the bore of the tubes or the specific gravity of the mercury are extremely small when the extreme fluctuations of the gauge readings ars noted, as will appear later.

METHODS.

The measurements for determining the capacity and efficiency of the pumps and the drop in pressure in the pipe line were usually made at the same time. The latter will be referrrd to later.

It was necessary to vary the mode of procedure somewhat in the different dredges, but, generally speaking, a Pitot tube was inserted in one of the suction pipes and in the discharge pipe. The tube in the discharge pipe was placed as far from the pump as possible, to avoid the effect of the disturbance caused by it. Owing to the fact that on none of the dredges is there a piece of pipe forming the suction pipe which has any considerable length between elbows or bends of various sorts it was difficult to so locate the tubes in the suction pipes as to be beyond the effects of these bends. For this reason the velocities obtained in the suction pipes have not been used in reducing results. A piezometer was introduced in the suction pipe usually within 4 or 5 feet of the pump flange.

Piezometers were introduced into the discharge pipes at various convenient points along its length. All piezometers were one-fourth-inch air cocks, screwed into holes tapped in the horizontal center of the pipes, and were connected by short rubber tubes to gauges, already described.

Drawings submitted herewith show the position of the pumps and arrangement of pipes on each dredge; also the points where piezometers and Pitot tubes were attached.

Pitot tubes were attached to differential gauges. Indicators were attached to each cylinder of the engines.

When all was ready and the engines running regularly, the gauges were read until ten sets of readings had been taken. During this time the speed of the pump, the steam pressure, and readings of the regular spring gauges attached to the suction and discharge pipes were observed and indicator cards taken. The time required for completing a set of observations was from five to ten minutes.

These observations were made at various speeds of the pumps. The limiting speeds were fixed on the one hand by a speed so low that experience in operation had shown it to be too slow for accomplishing the most work; on the other hand, the engines were run as fast as it was thought expedient to run them without risk of injury, and in each case faster than it would be feasible to operate them regularly. The mercury in the gauges connected with both the piezometers and Pitot tubes and on both the suction and discharge pipes fluctuated very rapidly and irregularly and through an interval extending sometimes over a space of 2 or 3 inches.

At first glance it seemed impossible to read the gauges with any degree of accuracy, but by watching them closely for several moments it was seen that there was a mean position near which the mercury came to rest at short intervals. In reading, each observation was made the mean of the fluctuations as nearly as could be judged, and as each side of the gauge was read from five to ten times for each observation, it is thought that the mean is probably correct within one-tenth of an inch.

In reducing the readings of gauges connected with piezometers a correction was applied, due to the height of the mercury on the pressure side of the gauge above or below the center of the pipes, or in other words, all piezometer pressures are reduced to the pressure at the center of the pipe. In the few cases where the impact and static pressure points of the Pitot tubes were attached to separate open gauges, the readings were also reduced to give pressures at the level of the point of the tube. Where, as was usually the case, the Pitot tube was attached to a differential gauge there was no correction to be applied on account of the position of the gauge above or below the point of the tube, as the same correction would apply to each side. There was a correction to be made due to the unbalanced column of water in one leg of the gauge of a height equal to the difference of the heights of the mercury in the two legs of the gauge, or in other words, the difference of readings of the height of the mercury in the two legs of the gauge, which was apparently the height due to the pressure being observed was reduced by the weight of an equal amount of water.

The mechanical or indicated horsepower developed by the engines was obtained from indicator cards taken on all the cylinders simultaneously in the usual manner. In obtaining the total head against which the pump worked the suction head was taken from piezometer readings in the suction pipe near the pump. Readings were taken in one suction pipe of a double suction pump only, it being assumed that the suction in either pipe would be the same. Some experiments made on the *Epsilon* showed this to be practically true.

In determining the delivery head it was thought that a gauge placed close to the pump might not give the true pressures, owing to the violent disturbance in the flow. For this reason piezometers were introduced in the straight pipe and some

distance from the pump. The platted line showing the fall in pressure in the pipe line referred to later on, was extended back to the flange of the pump, and this pressure corrected for any actual lift or fall in the pipe between the pump and first gauge was used for the delivery head.

If the areas of the suction pipe and discharge pipe, where measurements were made, were the same, the sum of the suction and delivery heads, as indicated by piezometers, would be the total head against which the pump was working.

This statement is contrary to the methods used in computing the work done by the pumps, in previous tests made of the dredges and published in the annual reports of the Mississippi River Commission, and may require some explanation. In these reports the method has been, in determining total head, to take the sum of the suction and discharge heads plus the velocity head in the discharge pipe.

The total energy imparted to the water is expended, outside the pump itself, as follows: Overcoming the entrance head, friction in the suction and discharge pipes, actual lift, and creating velocity. It is generally known that on a discharge pipe flowing under pressure a piezometer will give all the head except the velocity head. On a suction pipe, however, a piezometer gives the total head including the velocity head. This fact is quite clearly shown by Mr. M. White in the Journal of the Association of Engineering Societies, October, 1900, where he proposed to and did measure velocities by means of piezometers in the suction pipes.

Some measurements made in the suction pipe of the *Delta* also showed that this assumption concerning a piezometer in the suction pipe was true. If these facts concerning the piezometers are accepted, it is evident the sum of the piezometer readings in the suction and discharge pipes gives the total head, including the velocity head, and to add the velocity head to this sum, as was done in the computations mentioned above, would be crediting the pump with velocity head twice.

On all the dredges except the *Delta* the area of the suction pipes is somewhat greater than that of the discharge pipe, and the difference of the heads due to velocity in the suction and discharge pipes has been added to the sum of the suction and discharge piezometer readings. On the *Delta* conditions are reversed, and the difference in velocity heads was subtracted from the sum of the section and delivery heads. In other words, total head, H, may be expressed

$$H = h_D + \frac{V^2_D}{2g} + h_s - \frac{V^2_S}{2g}$$

Where h_D represents head on the discharge pipe and v_D the velocity at the same point, and h_S and v_S represent head and velocity in the suction pipe.

THE VELOCITY CURVE.

All measurements of velocity for determining capacity and efficiency were taken at the center of the pipes and it was necessary to determine the relation of center to mean velocity. This was done by "traversing," or measuring the velocity at 2-inch intervals across the diameter of the pipe. This was done on each of the pipes and at the points where velocity measurements were made. All were made on a vertical diameter except on the *Iota*, where traverses were made on both horizontal and vertical diameters.

As all bends in the pipes in the vicinity of the point of application of the Pitot tubes which would affect the form of the velocity curve are in a vertical plane, it is thought that the vertical velocity curve will give the correct mean velocity.

The results of the traverses were platted on cross-section paper, and the mean velocity of each curve obtained by dividing the pipe into 10 concentric rings of equal area and taking the mean of the velocities at the centers of these rings. These curves are appended hereto; each platted velocity is the mean of ten or more observations.

The mean value of $\frac{vm}{vc}$=0.8721 for all the dredges, and it was at first intended to use this value in reducing observations on all dredges. The application of this value to results gave efficiencies widely different in cases where there was every reason to believe that approximately the same conditions existed, and on further study it was noted that although there is a variation of the value of $\frac{vm}{vc}$ from 0.8008 on the *Delta* to 0.9711 on the *Gamma*, the variation between individual determinations on any one boat seldom varies more than 2 or 3 per cent, and it was decided to use, for reducing center to mean velocities on each dredge, the ratio as determined on that individual dredge.

This value does not seem to vary in proportion with the velocity, but is probably determined by local conditions on each dredge, the proximity of bends, etc. This is especially noticeable on the *Flad*, where traverses made in the discharge pipe inside the dredge, about 110 feet from the pump and with a straight pipe from the pump, with a mean velocity of 17.44 feet per second, gives a ratio of 0.788: while a traverse made in the pipe on the first ponton, about 40 feet beyond the short reversed bends at the stern of the dredge, with a mean velocity of 14.80 feet per second, gives a ratio of 0.910. These traverses were made at different times and at different speeds of the pump.

RESULTS.

A table is appended hereto, showing data and results on each dredge and at various speeds.

The efficiency of the pump and engine as a whole is given. Efforts were made to ascertain the mechanical efficiency of the engines by obtaining the horsepower required to run them without load. The results obtained were very unsatisfactory, owing to the difficulty of obtaining satisfactory cards. None of the automatic governors would hold the engines down to a normal speed without load, and to control the speed by throttling resulted in very poor cards.

From such cards as were obtained the frictional resistance varied from about 6 per cent on the *Gamma* to about 12 per cent on the *Iota*. It may be assumed, then, that the engine efficiences are from 90 per cent to 92 per cent, and the column of pump efficiency in the table of results has been computed on a basis of 92 per cent engine efficiency.

In general, the total head against which the pump was working has increased about in proportion with the revolutions. The efficiency, compared with speed, however, varies somewhat in different pumps. On the *Gamma* the variation in efficiency between speeds of from 142 to 167 revolutions is about 6½ per cent. On the *Epsilon* the efficiencies vary very little up to a speed of 170 revolutions, when they fall off rapidly with an increase of speed beyond that point. On the *Zeta* the efficiencies increase up to a speed of 170, then they fall off. On the *Iota* there is very little difference in efficiency over the range in speed covered; on the *Kappa* the efficiency falls off regularly as the speed increases from the lowest; on the *Flad*, however, at a speed only five revolutions less than on the *Kappa* the efficiency is about 7½ per cent less, increasing to about the same efficiency as the *Kappa* at the same speed, showing that the maximum efficiency on these dredges is attained at a speed of about 120 to 125 revolutions.

Comparing the maximum efficiencies with the peripheral velocity on the different pumps, the maximum on the *Gamma* is at a velocity of 50.27 feet per second; on the *Epsilon*, 50.58 feet; *Zeta*, 51.48; *Iota*, 54.98; *Kappa* and *Flad* 45 to 48 feet. Apparently the open-runner pumps require a slightly greater peripheral velocity to obtain the maximum efficiency than those with closed or shrouded runners, but the difference is slight. Generally speaking, it would seem that a peripheral velocity of about 48 to 50 feet per second will give the best results.

It must be borne in mind that these efficiencies were obtained while pumping water, and that the deductions may not be true when pumping sand. As velocity is the vehicle of transportation, it would seem that within certain limits the efficiency as a sand pump would increase with the velocity of discharge. It is hoped to procure additional data as to the proportion of sand carried at different velocities.

The following table shows the characteristic conditions accompanying the highest efficiencies of each pump:

Dredge.	Efficiency of pump and engine.	Total head (feet of water).	Peripheral velocity (feet per second).
	Per cent.		
Gamma	71.0	26.03	50.27
Delta	56.6	34.87	44.71
Epsilon	74.1	38.68	50.58
Zeta	73.4	50.15	51.48
Iota	69.1	48.29	54.98
Kappa	71.8	38.42	43.98
Flad	74.6	48.01	48.01
Mean		40.80	49.30

RELATION OF DISCHARGE TO LENGTH OF DISCHARGE PIPE.

Especial attention is invited to the mean velocities on the *Kappa*, with 240 feet of floating discharge pipe, and the *Flad*, which had 480 feet. For the range of speeds covered on each boat, from 115 to 133 revolutions, the mean total head varies from 42.05 feet on the *Kappa* to 41.65 feet on the *Flad;* while the mean velocity, however, is 21.13 and 16.75 feet per second, respectively. With a short discharge pipe a greater proportion of the work of the pump is converted into velocity, and, as before stated, as velocity is the only means of transportation, as short a discharge pipe as possible is desirable.

The same observations apply to the *Epsilon*, with 500 feet of floating pipe, and the *Zeta*, with 1,000 feet, though as these pumps are not exactly alike the comparison can not be as rigidly made.

This point illustrates the advantages possessed by the discharge pipes belonging to the *Iota*, *Kappa*, and *Flad*, which are mounted on pontons and entirely above the water surface and can be deflected to discharge on the side of the dredged cut, while it is impossible to deflect the pipe lines belonging to the older dredges to any great extent, owing to the fact that the pipes are partly submerged. It is therefore necessary to use sufficient discharge pipe to carry the material into deep water below the end of the cut.

EFFECT OF FORM OF SUCTION HEAD.

In order to ascertain what effect the suction head has on the efficiency of the pump, it was removed on the *Gamma* and *Epsilon* and measurements taken. Very little difference, if any, in results is noted, though on the *Gamma* the efficiency seems to be somewhat higher.

On all the dredges the area of the openings in the suction heads is considerably greater than the area of the suction pipes to which they are attached, and as they contain no abrupt bends affecting the line of flow of the entering water it is not thought that the form affects the velocity.

It is believed that a head built of such form and strength as to withstand the very severe usage to which it is subjected is much more important than one of a theoretical form. It is believed, on the other hand, that the area of the openings should not exceed the area of the suction pipe, in order that the same velocity may be maintained throughout.

GENERAL COMPARISON OF PUMPS.

The table of results shows mean efficiencies of engines and pumps at what may be called normal speeds, as follows:

Dredge.	Speed (revolutions per minute).	Mean efficiency.
		Per cent.
Gamma	142 to 167	66.6
Delta	122 to 124	56.1
Epsilon	158 to 171	73.2
Zeta	159 to 171	71.2
Iota	165 to 168	68.0
Kappa	120 to 133	69.4
Flad	120 to 131	71.8

Drawings submitted herewith show the general outlines of the pumps, with their characteristic features and dimensions.

Nearly all the pumps differ in more than one particular, and it is difficult to point with assurance to any one feature as being the one which affects the efficiency to the greatest extent.

The shrouded or closed runner on the *Epsilon* has a slightly higher efficiency than the *Zeta*, which has an open runner, but otherwise of the same form and dimensions. The clearance on the *Zeta's* runners was very small, not more than one-eighth to three-sixteenths of an inch. The *Epsilon's* runner also had very little clearance, not over one-eighth of an inch. The difference in efficiency is only 2 per cent, and this is less than the difference in efficiency of the *Kappa* and *Flad*, which are identically alike, of 2.4 per cent, so that very little additional efficiency can be claimed on account of the shrouded runner.

The fact that the *Epsilon's* efficiency is greater than that of the *Kappa* and *Flad*, while even the *Zeta*, with an open runner, equals them, seems to indicate that the form of the vortex chamber and tapering runner blades on these pumps possess some advantage. These runners also have narrower blades in proportion to their diameter and also in proportion to the discharge than the *Kappa* and *Flad*

The principal characteristic of the *Gamma*, in distinction from the *Epsilon*, *Zeta*, or *Iota*, is the long curved throat opening. In fact, the curve from the suction pipe to the tip of the runner is almost continuous. Compared with the *Zeta*, which has an open runner of the same diameter, a peripheral speed approximately the same, but with an efficiency of about 4½ per cent more, it does not appear that this long throat opening possesses any advantage, and it is certainly much more expensive to make and maintain. The runner on the *Gamma* has only four blades, instead of five, as on the other pump, and this may also affect the efficiency, and probably does.

The *Delta* has the only single-suction sand pump in use on any of the dredges. Its efficiency is so much lower than any of the others as to be quite marked. The drawing of the pump shows the characteristics and dimensions. It will be noted that, besides having the distinction of being the only single-suction pump, the blades of the runners are considerably wider than the others. Which feature is responsible for the low efficiency is hard to determine. It is possible that a higher speed would have developed a greater efficiency, but it is noted that the peripheral speed is as high as the most efficient speeds on the *Kappa*, although lower than the average. The total head against which the pump was working is not greatly different from a very efficient head on the *Epsilon*, though considerably below the heads showing the greatest efficiency on all pumps except the *Gamma*. The amount of water discharged per second is approximately the same as on the *Iota*, which discharged also through the same length of discharge pipe.

Referring, then, to paragraph 9 of the instructions, we are forced to the conclusion that the double-suction pump is the most efficient and that there are no indications that the flow of the two columns of water, entering from opposite sides, tends to reduce the efficiency of the pump. It must be observed, however, that owing to the construction of the runners these two columns of water do not meet till their direction has been changed so that they are approximately parallel with each other.

A table is submitted herewith showing the efficiencies of the pumps on the *Gamma*, *Delta*, *Epsilon*, and *Zeta*, as determined in the capacity tests made in 1897 and published in the report of the Chief of Engineers for 1898. In computing this table the mean of the measured quantities given in the above report have been taken and reductions of work done, as determined by the total head, made as explained above for the present report.

In this connection it should also be observed that the former report is in error, inasmuch as in computing the weight per cubic foot of the material pumped it is assumed to weigh 62½ pounds, while in fact the mixture of sand and water weighed nearly 75 pounds in some instances. The total head should also have been expressed in feet of material pumped instead of feet of water. However, as the weight of the material pumped would be increased in the same proportion as the head would be decreased, the results would remain the same. A comparison of this table with the results of the later test shows the efficiency of the *Gamma* to be almost exactly the same in each case. This is as it should be, as the pump and engines on the *Gamma* have not been changed in any way except to keep them in repair. It also indicates that the efficiency of a pump when handling sand or water is about the same, and that we can assume that the results obtained in pumping water, in the recent tests, can be applied to the pumps as sand dredges. The *Delta* shows the lowest efficiency again, though about 8½ per cent higher than obtained in 1902. The pump and engines are the same as originally made, and the only explanation of the difference lies in the fact, as intimated above, that the speed in the later tests was too low.

The runners on the *Epsilon* and *Zeta* were of the open pattern and had seven blades in 1897 and were alike. Apparently the larger number of blades possesses a higher efficiency than the present ones of five blades.

CONCLUSIONS.

From the foregoing we are led to submit the following conclusions:

That the most efficient speed for pumps of the size under consideration is one giving a peripheral velocity of about 50 feet per second, and that the most efficient head is 45 to 50 feet.

That the closed or shrouded runner, while being a necessity, to prevent abnormal

wear of the pump casings and runner, and thus increasing the clearance and decreasing the efficiency very rapidly, is also somewhat more efficient than an open runner, even with a small clearance.

That the tapering blades on the runner and curved vortex chambers are slightly more efficient, but that the additional expense of constructiou and maintenance is hardly warranted by the slight gain.

That the width of the blades on the *Kappa* and *Flad* is somewhat too great in proportion to the size of the pump.

That the area of the openings in the suction head, the area of the suction and discharge pipes should be approximately the same, in order that the velocity, which is the only vehicle of transportation, may be maintained at a nearly constant rate.

EFFICIENCY OF JET PUMPS OR SAND AGITATORS.

On the dredges under consideration there are installed two distinct types of jet pumps or agitators. On the *Gamma*, *Epsilon*, and *Zeta* there are single-suction centrifugal pumps with discharge pipes of 17 or 18 inches inside diameter; they deliver water to a pressure chamber just below the suction head provided with ten to thirteen jet openings 1¾ to 2 inches in diameter. The pumps are operated by a direct-connected vertical compound engine, condensing on the former and noncondensing on the two last-named dredges. In ordinary dredging operations in sand the pressure on the discharge pipe is from 10 to 15 pounds per square inch. This pressure is increased in material not readily loosened to 20 to 22 or 23 pounds, which is about the limiting pressure possible.

The dredges *Iota*, *Kappa*, and *Flad* are equipped with compound, duplex, condensing steam pumps, with 16-inch water plungers. They deliver water to a pressure chamber provided with 12 jet openings three-fourths of an inch in diameter, and are ordinarily operated at a pressure of 40 to 60 pounds per square inch. Their limit is about 80 or 90 pounds per square inch, and this pressure can not be maintained for any considerable length of time.

The table shows the results of observations made. The velocity of discharge, head, etc., for the centrifugal pumps was determined in the same manner as described under the main saud pumps. They show a combined efficiency of engine and pump of only about 41 per cent to 45 per cent, which seems quite low when compared with the larger pumps. This low efficiency can probably be explained by the fact that they have only a single suction, and there is therefore the friction of a thrust bearing to overcome, and that there is probably a greater clearance between runner and casing than on the large pumps.

In determining the efficiencies of the reciprocating pumps it was not possible to measure the velocities of flow in the discharge pipe, as the gauges and connections with the Pitot tubes were not designed for such high pressures.

The indicated horsepower was determined by indicators in the ordinary manner, but the discharge was computed from the displacement of the plungers and no allowance made for slip or leakage in the valves. This is not a very satisfactory determination, but it can readily be seen that even allowing for slip the mechanical efficiency would be considerably higher than the centrifugal pumps. It must be remembered, however, that the above efficiencies are based on indicated horsepower; and that the steam consumption with the reciprocating pump would probably be about 33 pounds per indicated horsepower hour, while with the compound engines we might expect at least as low as 24 pounds per indicated horsepower hour, so that the efficiencies when referred to the coal pile would not be so widely different.

The practical efficiency test would be the amount of work done in moving sand per indicated horsepower, and it is hoped that some tests along this line can be made later.

The writer believes the centrifugal jet pumps possess a very great advantage over the reciprocating steam pumps from a practical operating point of view, even should the latter show a greater steam economy for the same amount of work performed, for the following reasons:

No instance is recalled in the writer's experience, as superintendent for two dredging seasons, where a material was encountered by dredges equipped with centrifugal pumps that they were not as well able to handle as one equipped with the reciprocating pumps. It may be recalled that in 1898 the *Zeta* attempted to dredge through blue mud at Cherokee during a capacity test, and that her progress was very slow, the amount of material moved per hour being only about 600 yards. On the other hand, in 1902 the *Kappa* attempted to cut out a mud lump at Silver Top Bar, and an advance of only 50 feet per hour could be made,

cutting less than 3 feet in depth and in a very strong current. The jets in this material seemed to have no effect whatever.

The centrifugal jet pumps are extremely simple, contain no valves, and require no repairs, and practically no attention. The centrifugal pump on the *Gamma* ran for five seasons without even being taken apart for examination, and then the runner and casing showed no wear, the scale not even being worn off. The total repairs made consisted of a new brass bushing in the bearing of the shaft.

I think that the repairs of these pumps on the *Epsilon* and *Zeta* have not been more elaborate. The care and repair of the engines connected with them is not greater than that required for the steam end of the reciprocating pumps. The latter pumps, on the other hand, are a constant source of labor and expense in care and repair. It requires 120 feet of 1-inch square flax packing to pack the plungers on each of the pumps on the *Kappa*, *Flad*, or *Iota*, and this packing must be renewed at least once each season and more if a great deal of dredging is done.

The jet pumps on the *Delta* contain 260 hard-rubber valves, which required grinding on each side three times during the season of 1902, during which time five hundred and thirty-two hours were consumed in dredging, or, in other words, the valve surfaces required repair after a little less than four days' use. An entire set of new valves will be required for the next season. This example is an extreme one, though a large number of valves on the other pumps require renewal every year. The pumps require constant attention and care, keeping packing tight, etc.

THE LOSS OF HEAD DUE TO FRICTION IN DISCHARGE PIPE.

An inspection of the table of results shows that the delivery head is a very large proportion of the total head against which the pumps were working, amounting, on the *Gamma*, *Delta*, *Epsilon*, and *Zeta*, where discharge pipes are nearly horizontal from the pumps, from about 55 per cent to 60 per cent with 500 feet of floating pipe, from 70 per cent to 75 per cent with 1,000 feet. On the *Iota*, *Kappa*, and *Flad* this varies from about 60 per cent to 75 per cent, but on these dredges it includes, besides the loss by friction, an absolute lift of about 5 feet at the gooseneck at the stern of the dredge.

The loss in head was determined in the following manner: Holes were drilled in the suction and discharge pipes on the horizontal center and tapped for one-fourth inch pipe thread, and into these holes were screwed ordinary commercial one-fourth inch air cocks. The hole in these cocks at the screwed end is about one-fourth inch in diameter and is reduced through the plug and at the outlet end to about three-sixteenths of an inch. Short pieces of rubber tubing connected the cocks with gauges, already described, which rested on top of the pipes or in other convenient positions. Gauge readings were in each case corrected for the height of the gauge above or below the piezometer to give the true pressures at the center of the pipe.

The positions of the piezometers along the pipe lines are shown in the accompanying sketches.

Circumstances were such that it was impossible in most cases to go inside the pipes and ascertain if the ends of the cocks were exactly flush with the inside of the pipe, but from the known thickness of the pipe and length of thread on the shank of the cock they could be screwed in till they were approximately flush.

In connection with the comparison of the static pressure indicated by tube 9 and four piezometers placed on a vertical section and filed smooth with the inside of the pipe mentioned in the discussion of the Pitot tube four piezometers were placed in a vertical section 5 feet downstream from the first set. These were put in in the ordinary manner as described above, and each of the eight connected to open gauges and readings taken at the same time.

The mean of twenty observations of gauges connected to the four piezometers in the first set showed a pressure of 6.01, 6.05, 6.02, and 6.16 inches of mercury, respectively, a mean of 6.06, while the pressures indicated by the piezometers in the second set were 5.74, 5.35, 4.53, and 5.26 inches of mercury, respectively, a mean of 5.22. Correcting this for the drop in pressure in 5 feet of pipe, which in this case amounts to 0.25 inch of mercury, gives a means of 5.47, or 0.59 inch of mercury as the error. This error will always be in one direction and probably very nearly constant, so that relatively the indicated drop in pressure in a pipe line will not be far from correct.

The effect on the determination of the total head against which the pumps work would be to make it too small and the efficiency of the pumps too low. As the delivery head has been obtained in each case, as already explained, by a combination of piezometers on the discharge pipe, it is thought that the error on this account will not exceed 2 or 3 per cent.

It will be observed that the above observations on the first set of piezometers do

not agree as well as those mentioned in comparison with the Pitot tube, where each one was in turn connected with every other one through a differential gauge and all gave exactly the same pressure. The explanation is probably to be found in the fact that these latter measurements were made on open gauges and the variations in pressure are due to errors in gauge readings due to extreme fluctuations, as mentioned before, and especially as efforts were made to read these gauges at exactly the same instant, instead of, as was usually the case, reading the gauges over a given period of time and attempting to get readings of mean fluctuations. On the other hand, the differential gauge was very steady when attached to two piezometers. The fluctuations hardly extended over more than one-tenth of an inch, and three experienced observers were not able to detect any preponderence of pressure in favor of one side over the other.

Diagrams are submitted herewith on which the observed pressures on the discharge pipe at various points along its length have been platted for each dredge and at various speeds. These diagrams also show in dotted lines how the delivery head at the pump was obtained, as referred to under the efficiency of the pumps.

The *Gamma*, *Delta*, and *Zeta* show a fairly uniform drop from the stern of the dredge along the pipe line. On the *Gamma* and *Delta* the drop from the last gauge inside the dredge to the first one on the pipe line is not uniform with the rest of the line owing to a change in diameter of pipe inside and outside the dredge. The drop is also complicated inside the *Delta* by an absolute drop in the elevation of the pipe. On these dredges the pipe line was very nearly straight, and there was very little leakage at the connecting joints. The drop in pressure was not observed on the *Epsilon*, and the delivery head was obtained from observations on one gauge and extending the line of drop back, as indicated on the *Zeta*.

The *Iota*, *Kappa*, and *Flad* show interesting indications of the effect of short, abrupt bends on the loss of head. On each of these dredges there is a reversed bend at the stern of the dredge, the effect of which is shown in the drop between the last gauge on the dredge and the first one on the pontoons. This drop on the *Iota* is from 9.5 to 12 inches of mercury. This amount may be partly accounted for by an absolute lift of 5 feet, or 4.4 inches of mercury, and a drop of about 2.8 inches, due to the horizontal distance between gauges, leaving the loss of head due to the bends of from 2.3 to 4.8 inches of mercury, or from about 10 per cent to 16 per cent of the total delivery head.

On the *Kappa*, with a shorter discharge pipe, the effect of the short bends is more marked, the net loss being 7.1 inches, or 25 per cent of the total delivery head. On the *Flad* this loss amounts to about 2.4 inches, or a little less than 10 per cent.

There is also shown the increase in the suction due to the reversed curves in the suction pipe of the *Delta*. In this case the pipe on each side of the curves was about on the same level, as the bends are horizontal. The difference, then, of about 7 inches, or more than 50 per cent of the total suction head, is nearly all due to the bends. It does not seem to require any further argument as to the undesirability of short bends in the pipes.

The upper line on the diagram of the loss of pressure on the *Flad* shows the additional load caused by deflecting the pipes at each joint to the limit of their throw—about 20°—indicating that the loss due to slight bends, all in the same direction, is not great.

THE EFFECTIVE CROSS SECTION OF THE DISCHARGE PIPE AND PROPORTION OF SAND PUMPED.

Paragraph 7 of the resolution of the committee instructed that the velocity of discharge should be measured at several points in the cross section of the pipe, while pumping sand, to determine the effective cross section.

During the progress of the experiments already described we were not successful in determining the velocity of discharge while pumping sand, but were successful in determining the effective cross section, and several determinations were made of the proportion of sand carried.

The apparatus devised was as follows: A piece of 1-inch pipe, about 4 feet long, was fitted to go through one of the stuffing boxes already described as used with the Pitot tubes; the lower end of the pipe was bent to a right angle to face the current; the upper end of the pipe was fitted with two elbows, so arranged as to turn the opening downward; between the elbows was a gate valve for shutting off the flow of water. The fittings on the top of the pipe formed a handle, by means of which it could be manipulated. The stuffing box permitted the lower end of the pipe to be placed at any point along the vertical diameter at the point of application.

The lower end of the pipe being made to face the current and the valve being opened; the water would flow into the lower end and out at the upper end, as the vertical height of the pipe was much less than the head on the pipe at that point, and it was assumed that this stream of water would carry the same proportion of sand as was being carried in the discharge pipe at the point of the tube.

The water and sand issuing from the pipe were caught in iron buckets and tin cans and the proportional part of sand was measured. No attempt was made to determine the sediment or mud in suspension, but only that portion of the material which settled to the bottom of the receptacle at once or, say, within a minute. These measurements were made at intervals of 2 inches along the diameter. Three sets of traverses were made on the *Delta* while dredging at Peters Crossing and eight on the *Kappa* in the chute of Presidents Island, and tables are submitted herewith showing results.

The evidence that the whole cross section of the pipe is effective is quite conclusive, as in no case was there any evidence that any sand was resting on the bottom of the pipe, as the small pipe was shoved down to the bottom in every set of observations.

The indicated proportion of sand carried varies widely, and from the nature of the work it must be true that the actual proportion of sand being carried varies constantly. It can not be assumed that sand flows in a constant, uniform stream into the suction head, and in fact we know very well that it does not do so. It will therefore be necessary to take a large number of observations to determine with accuracy what proportion of sand is carried.

The results of the three sets of observations on the *Delta* have been platted and the mean of the eight sets also platted. Two of the sets on the *Delta* were taken dredging downstream; the mean amount shows 14.4 per cent sand, while the mean dredging upstream is 28.4 per cent sand. This is explained by the fact that in dredging downstream the only power available for moving the dredge is the force of the current. The dredge can not be forced into a bank of sand, as is possible in dredging upstream. The mean of all observations on the *Kappa* is 16.9 per cent of sand carried. The mean platted line of all results shows that the proportion generally increases from the top toward the bottom; though the table shows, in one instance at least, a greater proportion near the top than near the bottom. The proportion varies from 3 per cent to 46 per cent.

These measurements were made experimentally and near the close of the season, when very little time was available. They are not submitted as conclusive evidence, and it is hoped that more measurements can be made and that at the same time the velocity can be determined and thus determine the relation of velocity to sand-bearing capacity.

It will be noted that on the diagrams showing the loss of pressure on the *Iota*, *Kappa*, and *Flad* the point of zero pressure is at the last gauge, or about 30 feet from the end of the pipe, instead of at the end of the pipe, as would be expected. An explanation of this fact is not attempted by the writer at present, but that it was a fact on all three dredges is beyond doubt, as the observations were tested in various ways.

When the piezometer was connected to the ordinary mercury gauge it showed no pressure nor suction; the mercury would occasionally fluctuate from 0.05 inch above zero to an equal distance below. The mercury was then poured out of the gauge and it was filled with water to the height of the piezometer, and it still showed no pressure and no suction, though the water in the glass would occasionally fluctuate one-half inch above or below the zero.

The gauges were then removed and the air cock unscrewed from the hole in the pipe. This hole is one-half inch in diameter; but although it was on the horizontal center of a 32-inch pipe flowing entirely full and at a mean velocity of from 14 to 21 feet per second, no water came out except occasionally a few drops, which seemed to be caught on the downstream edge of the hole and thrown out. This observation was made on each of the three dredges mentioned above. How far from the end of the pipe the condition of no pressure extended was not determined. The fact is only true when the pipes were flowing entirely full at the ends. When on the *Iota* the speed of the pump was reduced to such an extent that the water filled the discharge pipe at the end to within 4 or 5 inches of the top, then the water in the gauge attached to the piezometer rose to approximately the height of the water in the pipe, as in an open channel.

After the capacity tests had been completed the Commission secured the services of Prof. W. B. Gregory, of Tulane University, to "examine the results obtained and suggest if any further measurements were desirable." No additional capacity tests were made, but some of the observations in connection with the Pitot tube,

mentioned above, were made at his suggestion. I am also indebted to him for suggestions as to the reduction of final results.

Asst. Engineer William Gerig rendered valuable assistance in making observations, and the final reductions and computations were nearly all done by him.

Very respectfully submitted.

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Assistant Engineer, Superintendent Dredging Operations.

Capt. Wm. B. Ladue,
Corps of Engineers, U. S. A.,
Secretary Mississippi River Commission.

Table of percentage of sand handled by Delta, Peters Crossing, November, 1902.

Distance from top of pipe.	1. Dredging upstream.	2. Dredging down-stream.	3. Dredging down-stream.
	Per cent.	*Per cent.*	*Per cent.*
4 inches	0.143	0.047	0.125
6 inches		.047	.111
8 inches	.177	.048	.117
10 inches		.058	.080
12 inches	.226	.089	.083
14 inches		.094	.200
16 inches	.276	.094	.175
18 inches	.267	.117	.176
20 inches	.243	.100	.141
22 inches	.286	.125	.235
24 inches	.304	.187	.187
26 inches	.321	.187	.141
28 inches	.358	.187	.219
30 inches	.371	.187	.250
32 inches			.250
33 inches (bottom)	.437	.250	
Mean	.284	.121	.166

Average of columns 1, 2, and 3, 0.190 per cent.

Table showing percentage of sand handled by Kappa, Presidents Island, November 28 and 29, 1902.

Distance from top of pipe.	1.	2.	3.	4.	5.	6.	7.	8.	Mean.
Top	0.031	0.067							
2 inches	.281	.193	0.125	0.125	0.088	0.100	0.063	0.030	0.126
4 inches	.255	.080	.140	.155	.125	.100	.063	.063	.123
6 inches	.271	.099	.125	.105	.125	.125	.083	.188	.140
8 inches	.207	.118	.155	.188	.150	.250	.125	.188	.172
10 inches	.247	.129	.155	.220	.125	.125	.125	.168	.162
12 inches	.050	.147	.188	.220	.188	.168	.083	.125	.146
14 inches	.110	.135	.188	.235	.214	.168	.188	.088	.166
16 inches	.073	.152	.155	.205	.175	.188	.167	.125	.155
18 inches	.214	.125	.250	.220	.171	.205	.083	.168	.167
20 inches	.147	.119	.188	.313	.168	.250	.083	.125	.174
22 inches	.131	.138	.188	.280	.168	.168	.125	.168	.171
24 inches	.177	.214	.280	.250	.188	.125	.125	.125	.185
26 inches	.130	.193	.280	.225	.179	.125	.125	.188	.181
28 inches	.137	.339		.219	.174	.168	.125	.125	.198
30 inches	.171	.467		.250	.168	.125	.125	.125	.204
Bottom, 32 inches				.250	.250	.188	.250	.250	.238
Mean	.165	.169	.186	.216	.168	.161	.121	.141	.169

The material was ordinary channel sand, possibly slightly finer than the average, pumped through 240 feet of 32-inch discharge pipe.

The measurements were made at the center of the second pontoon, or about 100 feet from the stern of the dredge.

Data and results of feed-water test of engines.

Description of main pumping engines.	Dredge Epsilon.		Dredge Kappa.	
Date of and duration of test	Dec. 9, 1902, 7 hours.		Dec. 17, 1902, 6 hours.	
Engines, pair of tandem compound direct connected	Noncondensing.		Condensing.	
Dimensions of engines:	*H. P.*	*L. P.*	*H. P.*	*L. P.*
Bore ... inches	16	26	15⅛	30
Stroke ... do	18	18	24	24
Diameter piston rods ... do	2½	3¼	$2\frac{9}{16}$	3½
Diameter tail rods ... do	----------	3¼	----------	3½
Average clearance, per cent of volume displaced by piston	----------	----------	----------	----------
Horsepower constant $\frac{1a}{3300}$:				
Crank end	0.008916	0.023756	0.010576	0.042257
Head end	0.009139	0.023910	0.010889	0.042527
Mean effective pressure starboard engine:				
Crank	71.04	15.32	61.60	15.90
Head	70.78	15.00	57.31	18.53
Mean effective pressure port engine:				
Crank	79.99	16.63	83.86	22.74
Head	79.82	16.58	86.41	24.18
Mean steam pressure at throttle valve	128.4	----------	150	----------
Mean vacuum on condenser	----------	----------	21.7	----------
Mean revolutions per minute	172.7	----------	129.8	----------
Mean indicated horsepower starboard engines	221.12	124.83	166.86	189.50
Mean indicated horsepower port engines	249.16	136.71	237.34	258.31
Mean total indicated horsepower per hour	731.82		852.01	
Total weight of water consumed, corrected for leaks	136,370.990		111,970.602	
Percentage of water in steam	$2\frac{1}{10}$		$2\frac{2}{10}$	
Total quantity of dry steam consumed	133.507.199		109,507.249	
Weight of dry steam used per hour	19,072.457		18,251.208	
Weight of dry steam per indicated horsepower per hour	26.06		21.42	

NOTE.—The above amount of steam used includes, on the *Kappa*, that used in the condenser, which is a compound duplex noncondensing jet condenser.

Results of boiler tests, 1902

	Dredge Gamma.	Dredge Delta.	Dredge Epsilon.	Dredge Zeta.
Date	Oct. 6, 1902.	Sept. 18, 1902.	Oct. 7, 1902.	Oct. 21, 1902.
Duration of trial	12 hours.	12 hours.	12 hours.	12 hours.
Type of boiler	3-5 Flue Miss.	3 Heine.	6-5 Flue Miss.	6-5 Flue Miss.
Grate surface ... square feet	61.75	107	154	154
Water-heating surface ... do	1,845.5	6,120	3,879	3,879
Ratio of grate to heating surface	1-29.8	1-57	1-25	1-25
Average steam pressure	135.2	148.6	124.1	133.4
Average temperature of feed water entering boilers	73.7	188.3	183.2	191
Temperature of stack gases	573	788.6	599	667
Percentage of moisture in steam	1.2	1.05	1.2	1.4
Total amount of coal consumed, pounds	18,651	29,718½	39,220	40,414
Total amount of refuse, dry, pounds	642	1,657½	2,999	2,650
Total amount of combustible	18,009	28,061	36,221	37,764
Coal consumed per hour	1,554	2,476.5	3,268.3	3,367.8
Coal consumed per hour per square foot, grate surface	25.16	23.1	21.2	21.8
Coal consumed per hour per horsepower	3.3	4.0	4.3	4.1
Total amount of water pumped into boilers and apparently evaporated, corrected for difference of level, etc	163,044	246,503	293,915	319,865
Water actually evaporated, corrected for quality of steam	161,087	243,915	290,388	315,387
Equivalent water evaporated into dry steam from and at 212°	191,693	261,965	312,167	336,833
Equivalent water evaporated into dry steam from and at 212° per hour	15,974	21,830	26,014	28,069
Equivalent water evaporated into dry steam from and at 212° per pound of coal	10.27	8.81	7.96	8.33
Equivalent water evaporated into dry steam from and at 212° per pound of combustible	10.64	9.34	8.61	8.92

Results of boiler tests, 1902—Continued.

	Dredge Gamma.	Dredge Delta.	Dredge Epsilon.	Dredge Zeta.
Equivalent water evaporated into dry steam from and at 212° per square foot of heating surface per hour	8.65	3.56	6.70	7.24
Equivalent water evaporated into dry steam from and at 212° per square foot of grate surface per hour	258.70	204.01	168.92	182.26
Commercial horsepower developed, based on the evaporation of 34½ pounds of water from and at 212° per hour	463	632.7	754	813.5

	Dredge Iota.	Dredge Kappa.	Dredge Flad.
Date	Sept. 17, 1902.	Oct. 18, 1902.	Oct. 3, 1902.
Duration of trial	9 hours.	12 hours.	9.083 hours.
Type of boiler	7-4 Flue Miss.	7-4 Flue Miss.	7-4 Flue Miss.
Grate surface ... square feet	157.5	157.5	157.5
Water heating surface ... do	3,769	3,830	3,830
Ratio of grate to heating surface	1-23.9	1-24.3	1-24.3
Average steam pressure	150	161	141
Average temperature of feed water entering boilers	168	161	171
Temperature of stack gases	681	502	678
Percentage of moisture in steam	0.75	1.45	2.5
Total amount of coal consumed ... pounds	21,189	35,535	29,183
Total amount of refuse, dry ... do	1,639	1,177	826
Total amount of combustible	19,550	34,358	28,357
Coal consumed per hour	2,354.3	2,877.9	3,212.9
Coal consumed per hour per square foot grate surface	14.9	18.2	20.4
Coal consumed per hour per horsepower	3.9	3.8	4.5
Total amount of water pumped into boilers and apparently evaporated, corrected for difference of level, etc	170,325	288,224	209 253
Water actually evaporated, corrected for quality of steam	169,048	284,045	204,022
Equivalent water evaporated into dry steam from and at 212°	185,108	313,444	222,568
Equivalent water evaporated into dry steam from and at 212° per hour	20,568	26,120	24,504
Equivalent water evaporated into dry steam from and at 212° per pound of coal	8.73	8.82	7.63
Equivalent water evaporated into dry steam from and at 212° per pound of combustible	9.47	9.12	7.85
Equivalent water evaporated into dry steam from and at 212° per square foot of heating surface per hour	5.46	6.78	6.40
Equivalent water evaporated into dry steam from and at 212° per square foot of grate surface per hour	130.59	165.84	155.58
Commercial horsepower developed, based on the evaporation of 34½ pounds of water from and at 212° per hour	5.96	7.57	710.2

TESTING DREDGES.

Data and results of main-pump tests, 1897.

Name of dredge.	Revolutions per minute.	Steam pressure.	Vacuum inches mercury.	Total indicated horsepower.	Suction pipes.				Discharge pipe.			
					Inside diameter.	Area.	Mean velocity per second.	Velocity head feet of water.	Inside diameter.	Area.	Mean velocity per second.	Velocity head feet of water.
		Lbs.			*Ins.*	*Sq. ft.*	*Feet.*		*Ins.*	*Sq. ft.*	*Feet.*	
Gamma	168.2	141.6	25.52	485.37	24¾	6.682	9.115	1.29	34	6.305	9.660	1.45
Delta	150.8	157.9	23.33	1,039.6	33¾	6.212	15.107	3.55	34¼	6.492	14.455	3.25
Epsilon	181.6	159.4	......	748.92	24¼	6.414	14.566	3.30	32¼	5.673	16.468	4.22
Zeta	181.4	146.2	......	686.94	24¼	6.414	14.343	3.20	32	5.585	16.472	4.22

TESTING DREDGES—continued.

Data and results of main-pump tests, 1897—Continued.

Name of dredge.	Discharge per second.	Discharge per second.	Heads feet of water.				Work done per second.	Efficiency of pump and engine.
			Suction.	Delivery.	Velocity.	Total.		
	Cu. ft.	*Pounds.*					*Ft. lbs.*	*Per cent.*
Gamma	60.907	3,806.69	9.38	37.0	0.16	46.54	177,163.35	66.4
Delta	93.845	5,865.31	18.62	45.54	−0.30	63.86	374,558.70	65.5
Epsilon	93.424	5,839.00	17.2	36.9	0.92	55.02	321,261.78	77.9
Zeta	91.996	5,749.75	11.52	35.1	1.02	47.64	273,918.09	72.5

NOTE.—These efficiencies are obtained by using the mean quantities measured, as given in Report of Chief Engineers, 1898, and reducing results by same methods as used in tests of 1902. in which 62.5 [illegible] of sand. Wrong

Data and results of main pump tests, 1902.

Name of dredge.	Revolutions per minute.	Steam pressure.	Vacuum mercury.	Total indicated horsepower.	Suction pipes.				Discharge pipe.	
					Inside diameter.	Area.	Mean velocity per second.	Velocity head of water.	Inside diameter.	Area.
		Lbs.	*Inches.*		*Inches.*	*Sq. feet.*	*Feet.*	*Feet.*	*In.*	*Sq. feet.*
Gamma	142	138	22.0	297.47	24⅝	6.682	12.982	2.62	34	6.305
	154	147	22	356.91	24⅝	6.682	13.680	2.90	34	6.305
	160	139	22	407.93	24⅝	6.682	14.395	3.22	34	6.305
	167	145	22.5	457.61	24⅝	6.682	15.007	3.50	34	6.305
	153	147	22.5	351.74	24⅝	6.682	13.600	2.87	34	6.305
	154	142	22.5	352.94	24⅝	6.682	13.870	2.99	34	6.305
Delta	122	160	23	737.39	33⅝	6.212	17.452	4.73	34½	6.492
	122	162	23	737.39	33⅝	6.212	16.965	4.47	34½	6.492
	124	165	23	784.15	33⅝	6.212	17.784	4.92	34½	6.492
Epsilon	158	144		587.50	24¼	6.414	17.554	4.79	32¼	5.673
	165	139		680.70	24¼	6.414	18.482	5.31	32¼	5.673
	166	141		687.92	24¼	6.414	18.356	5.23	32¼	5.673
	168	123		712.67	24¼	6.414	18.755	5.46	32¼	5.673
	171	140		765.97	24¼	6.414	19.088	5.66	32¼	5.673
	181	133		874.69	24¼	6.414	19.513	5.91	32¼	5.673
	181	133		867.41	24¼	6.414	19.441	5.76	32¼	5.673
Zeta	150	132		483.85	24¼	6.414	12.321	2.36	32	5.585
	159	134		571.15	24¼	6.414	13.067	2.65	32	5.585
	170	126		690.40	24¼	6.414	14.234	3.15	32	5.585
	171	125		712.06	24¼	6.414	14.304	3.18	32	5.585
	175	129		734.07	24¼	6.414	14.347	3.20	32	5.585
Iota	151	155	23	667.69	24	6.283	16.016	3.99	32 11/16	5.828
	165	150	24.8	817.83	24	6.283	16.622	4.29	32 11/16	5.828
	165	150	25	817.83	24	6.283	16.862	4.42	32 11/16	5.828
	165	150	25	817.83	24	6.283	16.751	4.36	32 11/16	5.828
	168	146	25.2	867.54	24	6.283	17.393	4.70	32 11/16	5.828
	168	146	25.2	867.54	24	6.283	17.393	4.70	32 11/16	5.828
Kappa	120	120	20	662.03	24¼	6.414	16.986	4.46	31⅞	5.541
	123	146	20.3	744.80	24¼	6.414	17.893	4.98	31⅞	5.541
	126	153	20	793.49	24¼	6.414	18.130	5.11	31⅞	5.541
	126	160	20	797.19	24¼	6.414	18.114	5.10	31⅞	5.541
	127	165	19.2	817.70	24¼	6.414	18.508	5.32	31⅞	5.541
	129	165	20	844.85	24¼	6.414	18.552	5.35	31⅞	5.541
	132	160	20	885.95	24¼	6.414	18.808	5.50	31⅞	5.541
	133	165	20	926.34	24¼	6.414	19.079	5.66	31⅞	5.541
Henry Flad	115	152	17	537.35	24¼	6.414	12.869	2.57	32	5.585
	115	139	15	521.53	24¼	6.414	12.869	2.57	32	5.585
	120	145	14.5	590.61	24¼	6.414	14.586	3.31	32	5.585
	126	123	20	634.98	24¼	6.414	14.796	3.40	32	5.585
	129	116	21	645 58	24¼	6.414	14.767	3.39	32	5.585
	129	117	21	644.31	24¼	6.414	14.896	3.45	32	5.585
	131	137	14	782.68	24¼	6.414	16.665	4.32	32	5.585
	128	137	18	701.29	24¼	6.414	15.249	3.62	32	5.585

Data and results of main-pump tests, 1902—Continued. ?

Name of dredge.	Discharge pipe.			Discharge per second.	Discharge per second.	Heads, feet of water.			
	Center velocity per second.	Mean velocity per second.	Velocity head of water.			Suction.	Delivery.	Velocity.	Total.
	Feet.	*Feet.*	*Feet.*	*Cu. feet.*	*Pounds.*				
Gamma	14.170	13.760	2.94	86.757	5,422.31	7.09	12.54	0.32	19.95
	14.930	14.499	3.27	91.416	5,713.50	7.90	13.90	.37	22.17
	15.710	15.256	3.62	96.189	6,011.81	8.75	15.36	.40	24.51
	16.379	15.905	3.93	100.281	6,867.56	9.29	16.31	.43	26.03
	14.843	14.414	3.23	90.880	5,680.00	8.24	13.80	.36	22.40
	15.138	14.701	3.36	92.690	5,793.12	8.32	13.90	.37	22.59
Delta	20.854	16.700	4.34	108.416	6,776.00	18.86	15.32	−.39	33.79
	20.272	16.234	4.10	105.391	6,586.94	19.72	15.52	−.37	34.87
	21.250	17.017	4.50	110.474	6,904.63	19.17	15.80	−.42	34.55
Epsilon	22.522	19.847	6.13	112.592	7,037.00	9.85	22.78	1.34	33.97
	23.715	20.897	6.78	118.549	7,409.31	10.91	24.97	1.47	37.35
	23.533	20.737	6.69	117.641	7,352.56	10.24	25.05	1.46	36.75
	24.064	21.205	6.99	120.296	7,518.50	11.50	25.65	1.53	38.68
	24.492	21.582	7.24	122.435	7,652.19	11.95	26.60	1.58	40.13
	25.036	22.062	7.53	125.158	7,822.38	12.49	28.86	1.62	42.97
	24.944	21.981	7.51	124.698	7,723.63	12.46	28.70	1.75	42.91
Zeta	17.443	14.150	3.08	79.028	4,939.25	5.85	29.81	.72	36.38
	18.500	15.007	3.50	83.814	5,238.38	6.47	33.75	.85	41.07
	20.150	16.347	4.15	91.298	5,706.12	7.72	39.60	1.00	48.32
	20.250	16.428	4.19	91.750	5,734.37	7.85	41.29	1.01	50.15
	20.310	16.477	4.22	92.024	5,751.50	8.28	40.16	1.02	49.46
Iota	18.600	17.181	4.59	100.131	6,258.19	5.74	33.01	.60	39.35
	19.400	17.920	4.99	104.438	6,527.38	6.71	38.70	.70	46.11
	19.680	18.179	5.14	105.947	6,621.69	6.68	38.75	.72	46.15
	19.550	18.059	5.07	105.248	6,578.00	6.77	38.75	.71	46.23
	20.300	18.751	5.47	109.281	6,830.06	6.79	40.30	.77	47.86
	20.300	18.751	5.47	109.281	6,830.06	7.22	40.30	.77	48.29
Kappa	22.436	19.663	6.01	108.953	6,809.56	10.70	26.19	1.53	38.42
	23.634	20.713	6.67	114.771	7,173.19	11.22	28.07	1.69	40.98
	23.947	20.987	6.85	116.289	7,268.06	11.72	28.44	1.74	41.90
	23.925	20.968	6.84	116.184	7,261.50	11.72	28.44	1.74	41.90
	24.446	21.424	7.13	118.710	7,419.38	11.90	28.81	1.81	42.52
	24.505	21.476	7.17	118.998	7,437.37	12.23	28.82	1.82	42.87
	24.843	21.772	7.37	120.639	7,539.94	12.75	29.05	1.87	43.67
	25.200	22.085	7.55	122.373	7,648.31	12.94	29.35	1.89	44.18
Henry Flad	18.742	14.780	3.40	82.546	5,159.13	9.19	26.91	.83	36.93
	18.742	14.780	3.40	82.546	5,159.13	9.19	26.91	.83	36.93
	21.243	16.752	4.36	93.560	5,847.50	9.78	29.25	1.05	40.08
	21.549	16.993	4.49	94.906	5,931.63	10.78	30.32	1.09	42.19
	21.505	16.959	4.47	94.716	5,919.75	11.30	29.87	1.08	42.25
	21.694	17.108	4.55	95.548	5,971.75	11.30	30.24	1.10	42.64
	24.270	19.139	5.69	106.891	6,689.69	11.64	35.00	1.37	48.01
	22.208	17.513	4.77	97.810	6,113.13	11.14	31.95	1.15	44.24

Data and results of main pump tests, 1902—Continued.

Name of dredge.	Work done per second.	Efficiency of pump and engine.	Efficiency of pump.	Diameter of runner.	Peripheral velocity per second.	$\frac{\sqrt{2gh}}{V_p}$	Length of suction pipe.	Length of discharge pipe.	Remarks.
	Foot lbs.	*Per ct.*	*P. ct.*	*In.*	*Feet.*				
Gamma	108,175.08	66.1	71.4	69	42.7519	0.8379	57.0	596	500-foot pontoons.
	126,668.29	64.5	69.7	69	46.3648	.8145	......	......	
	147,349.46	65.6	70.8	69	48.1712	.8242	......	......	
	178,762.58	71	76.7	69	50.2787	.8138	......	......	
	127,232.00	65.7	70.6	69	46.0637	.8241	......	......	Suction head removed.
	130,866.58	67.4	72.8	69	46.3648	.8221	......	......	Do.
Delta	228,961.04	56.4	60.9	84	44.7154	1.0436	87.6	620.5	500-foot pontoons.
	229,686.59	56.6	61.1	84	44.7154	1.0591	......	......	
	238,554.96	55.3	59.7	84	45.4485	1.0372	......	......	
Epsilon	239,046.89	73.9	79.8	69	47.5691	.9826	68.5	590	Do.
	276,737.72	73.9	79.8	69	49.6766	.9866	......	......	
	270,208.58	71.4	77.1	69	49.9776	.9728	......	......	
	290,815.58	74.1	80	69	50.5798	.9862	......	......	
	307,082.38	72.8	78.6	69	51.4830	.9869	......	......	
	336,127.67	69.8	75.4	69	54.4937	.9647	......	......	
	331,420.96	69.4	74.9	69	54.4937	.9641	......	......	
Zeta	179,689.91	67.5	72.9	69	45.1605	1.0711	68.5	1,092	1,000-foot pontoons
	215,140.26	68.4	73.9	69	47.8700	1.0737	......	......	
	275,719.72	72.6	78.4	69	51.1819	1.0892	......	......	
	287,578.65	73.4	79.3	69	51.4830	1.1032	......	......	
	284,469.19	70.4	76	69	52.6873	1.0705	......	......	
Iota	246,259.77	67	72.4	75	49.4147	1.0181	56	662	500-foot pontoons.
	300,977.49	66.9	72.2	75	53.9963	1.0086	......	......	
	305,590.99	67.9	73.3	75	53.9963	1.0090	......	......	
	304,100.94	67.6	73	75	53.9963	1.0099	......	......	
	326,886.67	68.5	74	75	54.978	1.0092	......	......	
	329,823.59	69.1	74.6	75	54.978	1.0137	......	......	
Kappa	261,623.29	71.8	77.5	84	43.9826	1.1302	54	392	240-foot pontoons.
	293,957.32	71.7	77.4	84	45.0820	1.1388	......	......	
	304,531.71	69.7	75.3	84	46.1815	1.1241	......	......	
	304,256.85	69.3	74.8	84	46.1815	1.1241	......	......	
	315,472.03	70.1	75.7	84	46.5480	1.1235	......	......	
	318,840.05	68.6	74.1	84	47.2811	1.1106	......	......	
	329,269.18	67.5	72.9	84	48.3806	1.0954	......	......	
	337,902.33	66.3	71.6	84	48.7472	1.0936	......	......	
Henry Flad	190,526.67	64.4	69.5	84	42.1498	1.1563	54	642	480-foot pontoons.
	190,526.67	66.4	71.7	84	42.1498	1.1563	......	......	
	234,367.80	72.1	77.9	84	43.9826	1.1544	......	......	
	250,255.47	71.6	77.3	84	46.1815	1.1280	......	......	
	250,109.44	70.4	76	84	47.2811	1.1026	......	......	
	254,635.42	71.8	77.5	84	47.2811	1.1076	......	......	
	321,172.02	74.6	80.6	84	48.0141	1.1574	......	......	Pipes deflected at each joint to full throw.
	270,444.87	70.1	75.7	84	46,9146	1.1370	......	......	

NOTE.—Efficiency of pumps alone are based on assumed efficiency of the engines of 92 per cent.

Data and results of jet pump tests, 1902.

CENTRIFUGAL PUMPS.

Name of dredge.	Revolutions per minute.	Steam pressure.	Vacuum inches, mercury.	Total indicated horsepower.	Heads feet of water.				Inside diameter of pipe.	Area of pipe.
					Suction.	Delivery.	Velocity.	Total.		
		Lbs.							*Inches.*	*Sq. ft.*
Gamma	144	138	22.0	72.50	0.92	33.30	0.45	34.67	$16\frac{7}{8}$	1.553
	150	148	20.0	82.40	1.10	35.84	.48	37.42	$16\frac{7}{8}$	1.553
	194	146	20.5	140.04	1.40	46.00	.70	48.10	$16\frac{7}{8}$	1.553
Epsilon	212	140	......	53.61	.06	23.73	.38	24.17	$17\frac{3}{4}$	1.718

Data and results of jet pump tests, 1902—Continued.

CENTRIFUGAL PUMPS—Continued.

Name of dredge.	Observed center velocity per second.	Mean velocity per second.	Discharge per second.	Discharge per second.	Work done per second.	Efficiency pump and engine.	Diameter of runner.	Peripheral velocity per second.	$\frac{\sqrt{2gh}}{V_p}$
	Feet.	*Feet.*	*Cu. ft.*	*Lbs.*	*Ft. lbs*	*Pr. ct.*	*Ins.*	*Feet.*	
Gamma	5.580	5.368	8,337	521.06	18,065.15	45.3	69	43.3541	1.089
	5.790	5.570	8,650	540.63	20,230.37	44.6	69	45.1605	1.086
	7.000	6.734	10,458	653.63	31,439.60	40.8	69	58.4076	.952
Epsilon	5.591	4.926	8,463	528.94	12,784.48	43.4	49	45.3259	.870

RECIPROCATING PUMPS.

Name of dredge.	Revolutions per minute.	Steam pressure.	Vacuum inches, mercury.	Total indicated horse power.	Delivery head.	Delivery head feet of water.	Distance from delivery gauge to water.	Total head feet of water.
		Lbs.			*Lbs.*			
Iota[a]	18	155	29	41.24	55	126.72	8	134.72
	20	155	29	54.23	67	154.37	8	162.37
Kappa	25½	161	18¾	32.48	20	46.08	8	54.08
	26	158	18¾	35.53	23	52.99	8	60.99
	29	152	18¾	61.48	47	108.29	8	116.29
	40	139	18½	102.10	62	142.85	8	150.85
	44	160	19	147.19	80	184.32	8	192.32
	45	150	19	156.50	85	195.84	8	203.84
	46	160	19	160.50	92	211.97	8	219.97
Flad	27	135	21½	42.64	40	92.16	8	100.16
	27	135	21½	48.66	40	92.16	8	100.16
	33	136	21½	80.92	70	161.28	8	169.28
	34	138	12	94.94	70	161.28	8	169.28
	38	135	21½	107.40	80	184.32	8	192.32
	38	138	12	115.46	80	184.32	8	192.32

Name of dredge.	Diameter of plunger.	Displacement of both plungers per stroke.	Water displaced per minute.	Water displaced per minute.	Work done per minute.	Work done per second.	Efficiency pump and engine.
	Inches.	*Sq. ft.*	*Cu. feet.*	*Pounds.*	*Ft. pounds.*	*Ft. lbs.*	*Per cent.*
Iota	16	5.510	123.975	7,748.44	1,043,869.84	17,397.83	76.7
	16	5.510	137.750	8,609.38	1,397,905.03	23,298.42	77.9
Kappa	16	5.510	175.631	10,976.94	593,632.92	9,893.88	55.4
	16	5.510	179.075	11,192.19	682,611.67	11,376.86	58.2
	16	5.510	199.737	12,483.56	1,451,713.19	24,195.22	71.5
	16	5.510	275.500	17,218.75	2,597,448.44	43,290.81	77.1
	16	5.510	303.050	18,940.62	3,642,660.04	60,711.00	75.0
	16	5.510	309.938	19,371.12	3,948,609.10	65,810.15	76.5
	16	5.510	316.825	19,801.56	4,355,749.15	72,595.97	82.2
Flad	16	5.510	185.976	11,623.50	1,164,209.76	19,403.50	82.7
	16	5.510	185.976	11,623.50	1,164,209.76	19,403.50	72.5
	16	5.510	227.304	14,206.50	2,404,876.32	40,081.27	90.0
	16	5.510	234.192	14,637.00	2,477,751.36	41,295.86	79.1
	16	5.510	261.744	16,359.00	3,146,162.88	52,436.05	88.8
	16	5.510	261.744	16,359.00	3,146,162.88	52,436.05	82.6

[a] Work done computed from displacement of plungers; no allowance for slip made.

LETTER OF ASST. ENGINEER F. B. MALTBY.

MEMPHIS, TENN., *June 17, 1903.*

CAPTAIN: As the Commission at its coming meeting may be interested in knowing the results of my recent tests with a 7-bladed runner, made on the *Zeta*, I have the honor to inclose herewith a table of results in the same form as that submitted with my report on testing.

As far as possible the same conditions were obtained in both tests. The same instruments for measuring were used and the same amount of discharge pipe was attached.

The mean efficiency with the 5-bladed runner at a speed of 170 to 175 revolutions is 72.1 per cent, while the mean efficiency with the 7-bladed runner at a speed of 168 to 175 revolutions is 73.1 per cent—no very great difference.

The diameter at the outside tip of the blades and at the inside shoulder was the same in each case. The amount of clearance, less than one-eighth inch, was also about the same.

The total amount of work done at the same speed is about 1 per cent less with the 7-bladed runner than with the 5, though with an efficiency of about 1 per cent higher.

Respectfully submitted.

F. B. MALTBY, *Assistant Engineer.*

Capt. WM. B. LADUE,
Corps of Engineers, U. S. A., Secretary Mississippi River Commission.

Data and results—Dredge Zeta with 7-blade runner.

[May 22, 1903. 1,000 feet of discharge pipe. $\frac{Vm}{Vc}$=0.8099 used.]

	Revolutions per minute.	Steam pressure.	Total indicated horsepower.	Suction pipes (2). Inside diameter.	Suction pipes (2). Area.	Suction pipes (2). Mean velocity per second.	Suction pipes (2). Velocity head of water.	Discharge pipe. Inside diameter.	Discharge pipe. Area.	Discharge pipe. Center velocity per second.
		Pounds.		*Inches.*	*Sq. feet.*	*Feet.*	*Feet.*	*Inches.*	*Sq. feet.*	*Feet.*
11.25 A	180	135	-----	24¼	6.414	-----	-----	32	5.585	-----
11.38 A	180	135	764.52	24¼	6.414	15.235	3.61	32	5.585	21.603
11.55 A	180	135	770.15	24¼	6.414	15.643	3.80	32	5.585	22.182
1.15 P	168	125	639.97	24¼	6.414	14.769	3.39	32	5.585	20.942
1.22 P	170	133	665.83	24¼	6.414	14.854	3.43	32	5.585	21.063
1.39 P	162	132	574.26	24¼	6.414	-----	-----	32	5.585	-----
1.45 P	161	132	569.33	24¼	6.414	14.017	3.05	32	5.585	19.875
1.57 P	180	137	754.83	24¼	6.414	15.294	3.64	32	5.585	21.687
2.06 P	175	128	703.53	24¼	6.414	15.093	3.54	32	5.585	21.402

	Discharge pipe. Mean velocity per second.	Discharge pipe. Velocity head of water.	Discharge per second.	Discharge per second.	Heads, feet of water. Suction.	Heads, feet of water. Delivery.	Heads, feet of water. Velocity.	Heads, feet of water. Total.	Work done per second.	Efficiency of pump and engine.	Diameter of runner.	Peripheral velocity per second.
	Feet.	*Feet.*	*Cub. ft.*	*Pounds.*					*Foot-lbs.*	*Pr. ct.*	*In.*	*Feet.*
11.25 A	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	69	54.192
11.38 A	17.496	4.76	97.715	6,107.19	6.90	40.73	1.15	48.78	297,908.1	70.85	69	54.192
11.55 A	17.965	5.02	100.334	6,270.87	7.13	41.83	1.22	50.18	314,672.4	74.28	69	54.192
1.15 P	16.961	4.47	94.727	5,920.44	6.13	37.35	1.08	44.56	263,814.2	75.01	69	50.579
1.22 P	17.059	4.52	95.275	5,954.69	6.28	38.22	1.09	45.59	271,474.3	74.14	69	51.1819
1.39 P	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	69	48.7728
1.45 P	16.097	4.03	89.902	5,618.88	5.73	34.46	0.98	41.17	231,329.3	73.87	69	47.4115
1.57 P	17.564	4.80	98.095	6,130.94	6.86	41.03	1.16	49.05	300,721.2	70.79	69	54.192
2.06 P	17.333	4.67	96.805	6,050.31	6.56	39.17	1.13	46.86	283,517.5	73.27	69	52.566

APPENDIX 1 G.

REPORT OF PROF. W. B. GREGORY ON EFFICIENCY TESTS OF HYDRAULIC DREDGES.

TULANE UNIVERSITY OF LOUISIANA,
New Orleans. La., May 18, 1903.

CAPTAIN: In accordance with instructions conveyed in a letter, under date of February 24, 1903, from Capt. G. P. Howell, I hereby beg leave to submit the following report:

The general outline of dredge tests required by the committee on dredges and dredging, under resolution of Commission, is given in the report of Mr. F. B. Maltby, United States assistant engineer, superintendent of dredging operations. In the main they embody the suggestions conveyed by the writer in a letter to Maj. B. M. Harrod, under date of April 15, 1902, but contain many other instructions as to details of work.

The writer can not refrain from expressing his admiration for the thorough, capable, and painstaking way in which these instructions were carried out, the tests made and reported.

Boiler and engine tests were not taken up in the letter, as it was understood that the tests of main pumps were considered of the greatest importance, and as methods of conducting tests of boilers and engines were well known, standard rules for such tests having been recommended by the American Society of Mechanical Engineers.

Early in November, Capt. G. P. Howell, Corps of Engineers, U. S. Army, then secretary of the Mississippi River Commission, and Major Harrod called on the writer at Tulane University and the former engaged his services as consulting engineer in these tests, informing him that many tests had already been made and asking him to visit the dredging fleet at its winter quarters near Memphis to examine data, confer with Mr. Maltby as to the desirability of further tests, and offer such suggestions as circumstances might demand.

According to instructions, writer left New Orleans on the evening of November 19, 1902, arriving at Memphis the next day. He returned to New Orleans November 26.

During this first visit much data were examined and ways were suggested to obtain constants for the various Pitot tubes and to establish what was believed to be a fact, namely, that tubes Nos. 8 and 9 had a constant equal to unity in the equation $v=c\sqrt{2gh}$.

Again, on December 6, writer left New Orleans for Memphis, remaining at the fleet until December 23, when he returned home.

During the second visit data obtained since first visit were examined, and, as they were not conclusive, further tests were conducted, in which the writer participated. These tests established the fact that the formula $v=\sqrt{2gh}$ was correct for tubes Nos. 8 and 9, and included tests of engines of dredges *Epsilon*, *Iota*, and *Kappa*. Results of tests will be discussed in the order given in the general description of tests by Mr. Maltby.

BOILERS.

These tests were undertaken to give data for ordinary, everyday running. This they do very well, but they do not contain all the information necessary for a satisfactory analysis of results, and so only the most general conclusions can be drawn.

Two kinds of boilers were tested, viz, the water-tube and a kind of horizontal fire-tubular boiler, which is known as the Mississippi River boiler. One peculiarity of the furnace in the latter type of boiler is the small distance between top of grate bars and bottom of boiler shells, which is much less than in stationary boiler practice. In the furnace of the *Gamma*, for instance, the distance from grate to bottom of boilers is 17 inches at the front, while at rear end of grate it is only 15 inches. In stationary boilers 20 to 24 inches, or even more, is considered good practice, the idea being to allow space enough for perfect combustion before the hot gases come in contact with the comparatively cool surfaces of boilers.

The flame bed is also high behind the bridge wall. It appears that the gases in passing backward along the surfaces of those boilers are compelled to pass between and around the sides, which may account for the high efficiency of heating surfaces.

According to Kent, for a fuel completely burned in the furnace and a given steam pressure, the evaporation per pound of combustible depends on several quantities, viz: The heating value of the coal; the difference of temperature between boiler and air; the weight of flue gas per pound of combustible, which in turn depends on the force of draft and the condition of the fire; rate of driving and radiation. It also depends on the manner in which gases pass through furnace, dead spaces, eddies, etc.

The amount of draft is one of the most important factors. The draft in all these tests was "good," but it was not measured. No analysis was made of the flue gases, so we do not know how much air, above the quantity needed to supply oxygen, was furnished, or if any.

The moisture in the coal was not determined; as coal was kept in open barges the amount of moisture contained is difficult to estimate.

Attention is called to these facts merely to show that a scientific analysis of results is impossible. However, one point is obvious: it is that in these tests the capacity of the Mississippi River boiler is far superior to that of the water-tube boiler on the basis of square feet of heating surface.

For each pound of water evaporated per hour from and at 212° the average heating surface of all tested was 0.148 square foot for the former and 0.28 for the latter. While the average heating surface per horsepower was 5.11 square feet for the fire-tube boilers, it was 9.69 square feet for the water-tube.

The average equivalent evaporation per pound of combustible from and at 212° for the former was 9.1, while for the latter it was 9.32. For the boilers of the *Gamma* it was 10.64.

Great capacity is obtained, especially in the Mississippi River boilers, at a sacrifice of economy, although comparison shows the economy of the fire tube to be nearly equal to, in the average case, and in one case to greatly exceed, that of the water-tube type.

Mr. Maltby has pointed out the gain to be had from the use of a feed-water heater; the point is well taken.

MAIN ENGINES.

In order to study the results of the tests of the engines of the *Epsilon* and *Kappa* more carefully, a set of cards from each test has been selected and combined. Separate cards from each test have been combined for the starboard and port engines, using mean volumes of high and low pressures, as shown by average of crank and head ends in each case. The particular set of cards to combine were chosen because of their close agreement in mean effective pressures and in number of revolutions per minute with the average of all cards taken during the test.

The clearances of the engines of the *Kappa* were obtained from the cards, assuming hyperbolic expansion. The values obtained were mean clearance, high-pressure cylinder, 11.6 per cent; low-pressure cylinder, 5.6 per cent.

It was impossible to use this method on the cards from the engine of the *Epsilon* as they were taken at an average of 172.7 revolutions per minute and showed many irregularities, which were absent from those of the *Kappa*, which were taken at an average of 129.8 revolutions per minute. The mean clearance of the high-pressure cylinders of the *Epsilon* were assumed to be 11 per cent and the low pressure 6 per cent.

From known clearances of other engines of the same make and approximately of the same size, it is believed that these values are not far from the truth.

A glance at these combined cards is sufficient to convince one of the necessary loss in the noncondensing engine. There is a large "drop" in pressure at high pressure release and a low ratio of expansion.

The indicator springs were calibrated by the writer and found to be within the limit of probable error of observation.

The consumption of steam per horsepower hour, 26.06 for the noncondensing and 21.42 for the condensing engine, showed a balance of 4.64 pounds in favor of the condensing engine. It is necessary to add about 21.7 per cent to the water rate of the condensing engine to obtain that of the noncondensing.

The results correspond to carefully conducted tests made elsewhere, and are also such as theory would lead us to expect when all the various losses are considered.

As the greater part of all the steam generated in the boilers is used in the main engines, the coal bill can evidently be greatly reduced by using feed-water heaters and condensing engines.

MAIN PUMPS.

The various problems presented by the main pumps are of the greatest interest.

METHODS OF MEASURING DISCHARGE VELOCITIES.

The method used to measure the velocity in discharge pipe is believed to be practically the only method that could be successfully employed in this case. Other methods were suggested. The Venturi meter is known to be an accurate instrument, capable of giving results within 3 per cent of the truth, and possibly closer average results; but unless an abnormally large meter could be used, the loss of head at the throat, for velocities such as were to be measured, would be excessive, and the pressure head necessary to overcome this loss would give abnormal conditions for the pumps. Its weight would necessarily be great, and after placing it on a suitable barge it would have to be attached successively to pipes of different diameters and at varying heights from the surface of the water, as the several pumps were tested.

A weir might have been constructed, but the uncertain stages of the river made it doubtful how long it would have been of service.

A measuring barge might have been used in which water could be pumped for a known length of time.

The Pitot-tube method is believed to be as accurate as any of the ways suggested above, and has the great advantage of ease of application without materially changing the normal condition of discharge.

The Pitot tubes Nos. 8 and 9, when calibrated in running water by means of floats, showed an average constant of unity in the formula $v=c\sqrt{2gh}$ within one-third of 1 per cent.

These two tubes under very different static pressures showed average velocity readings which differed by only one-tenth of 1 per cent. The difference in static pressure amounted to about 17.7 feet of water. One tube was used where the static pressure was nearly five times as great as that at the section where the other tube was used; yet, as already stated, results agreed within one-tenth of 1 per cent.

As these results were obtained from 170 observations the agreement is not accidental. This experiment shows that the results are not affected by a difference of static pressure. Other data are given in the report of Mr. Maltby illustrating this point still further.

It was also shown that the pressure obtained by means of a piezometer at the outside of a straight pipe in which water is flowing is precisely the pressure of the entire cross section of the pipe, corrected, of course, by height of point considered; and that it agreed exactly with the static side of a properly constructed Pitot tube at that section. That in a straight pipe a Pitot tube can be used with impact opening only, and static pressure obtained by means of a carefully inserted piezometer at outside of pipe, is thus clearly shown.

From the above facts it is evident that the behavior of a correctly designed Pitot tube, when used in an open canal or in a discharge pipe under pressure, is practically the same. There was no suction action on the static side of the tube, and the impact reading was not affected by great difference of static pressure. It therefore follows that the constant of tubes Nos. 8 and 9 must be unity.

In making a traverse of a pipe considerable time was required; as much as an hour or even more was sometimes used, consequently, irregularities appear, due to the practical impossibility of keeping conditions of speed of engines and pumps absolutely constant, as well as to other causes.

The pressure throughout the section of a straight pipe in which water is flowing, corrected for difference of height of the points examined, was shown to be a constant quantity, as already stated.

A great mass of data on this point was collected. The velocity parallel to axis of pipe was found to be very much greater at the center than at the surface. A platting of many results of the traverses in the discharge pipe of the *Epsilon* shows the mean-velocity curve to be an ellipse, and that the velocity of the entire cross section can be represented by an ellipsoid of revolution. This agrees with results found, at much lower velocities, by Messrs. Williams, Hubbell, and Fenkell in their Detroit experiments.

As pointed out by Mr. Maltby, there was some variation in the traverses in the general discharge pipes due to local irregularities, and so the ratio of center to mean velocity was not a constant quantity.

In some cases the center velocity was over 25 feet per second, while that near the pipe was as low as 18 feet per second, parallel to the axis of the pipe.

Bernouilli's theorem states that the sum of static and velocity heads in any two sections of the same pipe are equal when corrected for the loss of head due to friction between the two sections.

If it is understood to mean that the sum of the velocity and static heads at all points in a given cross section are equal, it can only be true when the absolute velocity of the various particles of water is considered and not the velocity parallel to axis of pipe. The particles of water in the outer part of the moving mass encounter friction with the walls of the pipe and are retarded; complicated cross currents are undoubtedly set up. Particles in the outer part of the moving stream exert an influence on their neighbors, and this extends to the center of the pipe, where the disturbances are least. The velocities of particles relative to each other are thus greatest at the outside and least at the center. If it were possible to measure these absolute velocities and so to find the velocity head corresponding to them. it is believed that Bernouilli's theorem would be true for all points in a cross section.

METHOD OF COMPUTING TOTAL WORK OF PUMPS.

The tests of 1897, made while pumping sand, have been reworked by Mr. Maltby and form a part of his report. These tests as originally published are incorrect, as they credit pumps with the sum of head shown by discharge pressure minus the negative head shown by suction pressure, which includes the velocity head in suction pipe and the computed velocity head in discharge pipe. Now, if suction and discharge pipes were of the same cross section, the method used in computing tests as reported gives twice the velocity head, and is therefore in error by the amount of the velocity head. As the suction and discharge pipes are usually unequal in sectional area a correction must be made, due to the difference of velocity head at these two sections. This point will be taken up again later.

Another point is misleading in the report of tests of 1897, although the actual error involved is very small. The various heads are always stated in feet of water. As a matter of fact a mixture of sand and water was pumped, and the heads ought to be measured in feet of liquid pumped—that is, the mixture.

In some cases instead of 62.5 pounds per cubic foot the mixture weighed between 70 and 75 pounds per cubic foot.

In any case the work done by a pump is equal to $Q\times W\times H$, where—

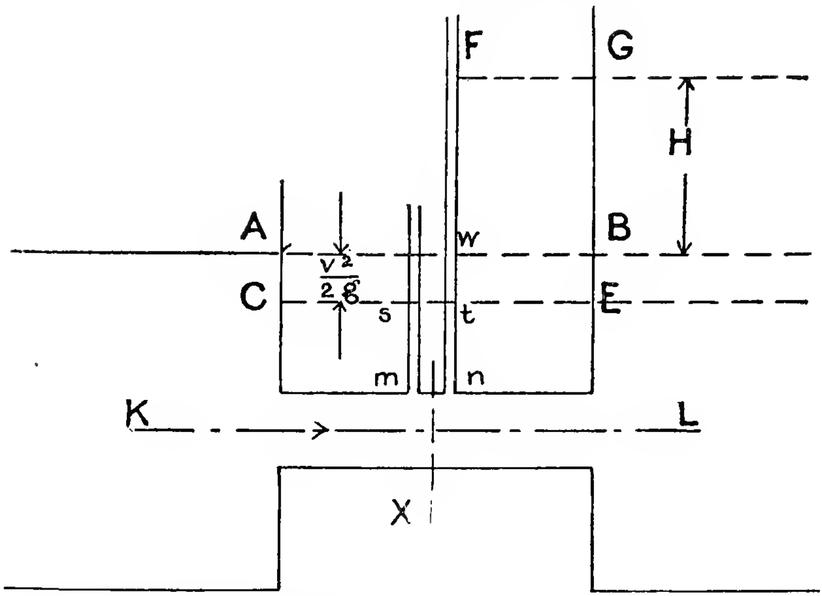
Q=volume pumped per unit of time.
W=weight per unit volume of liquid pumped.
H=total head$=h_d-h_s+h'+h''$.
h_d=discharge head, obtained by observing the pressure p_d, at some point in the discharge pipe, then $h=\frac{p_d}{W}$.
$h_s=\frac{p_s}{W}$=suction head, obtained at some point in suction pipe by observing the pressure p_s. It includes the velocity head, as will be shown.
h'=difference between velocity heads in discharge and suction pipes.
h''=difference in level of the two points where discharge and suction pressures are measured.

$$\text{Total work}=Q\times W\times(h_d-h_s+h'+h'')=Q\times W\times\left(\frac{p_d}{W}-\frac{p_s}{W}+h'+h''\right).$$

It will be seen that nearly all the total head is included in the first two terms of the quantity within the parentheses and that the error arising from computing results when pumping sand due to using the weight of a unit volume of water instead of the weight of unit volume of the mixture only affects the last two terms. In the case of h' it is small enough to neglect and probably so in the case of h''.

It is to be noted that h_s is always negative in these tests, but both h' and h'' may be positive, negative, or zero, depending on conditions in individual cases. Very little error will therefore be made by using W as the weight per unit volume of water instead of the weight of mixture pumped, and the method of computing results is greatly simplified.

It will now be shown that the suction pressure includes the velocity head in suction pipe where pressure is measured.



Referring to figure, let two tanks K and L be connected by a pipe of uniform cross section in which water flows with a constant velocity v when the level of the water in the tank K is at A. Suppose a suitable supply provided for tank K or that the water level is always kept at A; also an outflow provided for tank L, so that water may be discharged into L and a desired height always maintained.

Now, neglecting losses due to friction in pipe at entrance, etc., the "hydraulic gradient" would take the position of line CE, and this would be the level of the water in the tank L to maintain continuous flow at a velocity of v feet per second. $AC=BE=\frac{v^2}{2g}$. Let two piezometers be placed in pipe at m and n; the level in these piezometers would be s, t.

Suppose at section X we cut pipe and introduce a pump, which will always maintain the uniform velocity v in the pipe and also deliver the water into tank L at a level AB. The level of water in discharge tank will then just equal that in suction tank, and the work of the pump will clearly be to produce the velocity head.

At the left of section X the conditions are unchanged from those of the first case taken, for the constant velocity v is maintained.

The height of the water in piezometer m will therefore be at s. On the right of section X the level of water in tank L is raised to B, and therefore the water in piezometer n will stand at w.

The difference between the readings of the two piezometers will thus be seen to include the velocity head; in fact, in this case would be just equal to the velocity head.

Again, if water is pumped into tank L and the level in tank is raised to G, while the same velocity v is maintained in suction and discharge pipes, the level of water in piezometer n would be at F, while in the piezometer m it would still stand at s; the difference between these two readings would plainly be $EG=\frac{v^2}{2g}+H$ and is seen to include the velocity head.

The simplest possible case has been taken to prove this point; the proof still holds, however, if we consider all losses in the pipe.

It will be seen that if section at m differs in size from that at n, the hydraulic gradient will not be a straight line; but in any case the velocity head which is included is that at m, i. e., the velocity head corresponding to the velocity in suction pipe. The velocity head in discharge pipe will not be shown by piezometer, but may be computed. If it is more than that in suction pipe the difference must be added to suction and discharge heads; that is, h' is positive; if less, h' is negative; in the equation $H=h_d-h_s+h'+h''$.

The correction for difference of level h'', at which discharge and suction pressures are obtained, has been already referred to.

Then, knowing discharge head, suction head, the difference (if any) in velocity heads in discharge and suction pipes, the correction due to difference of level, and the proper sign for each and all of these quantities, we are able to compute the total head H.

From the mean velocity in discharge pipe the quantity pumped is known, and we thus have all quantities necessary to determine the total energy given by pump to water as accounted for between a point in suction pipe and another point in discharge pipe. One possible error arises from the assumption of the mechanical efficiency of all the engines as 92 per cent.

The cards taken on the *Gamma* showed the friction to be about 6 per cent, while on the *Iota* it was about 12 per cent. Any engineer who has tried to obtain the friction cards from an engine realizes how unsatisfactory the results usually are. In this case they were so much so that it was decided to use a constant quantity for all. It is not improbable that a small error was introduced here, which, if it could be known, would slightly change results.

EFFICIENCIES OF PUMPS.

When an attempt is made to find the reason of high efficiency in one case or low efficiency in the other of the main pumps the greatest difficulties are encountered.

Hydraulic problems are very complex and must be attacked largely from the experimental side. An example will illustrate this. Only three years ago it was maintained by some engineers of the highest standing in America, in discussing the Pitot tube, that velocity head and static head were not mutually convertible. The admirable researches of Mr. W. M. White settled this point.

In a centrifugal pump with many blades, and where the path of the water is therefore well circumscribed, it would be possible to take a series of tests, and, knowing the quantity of water pumped, angular velocity of impeller, shape of blades and of pump casing, to analyze those results with more or less accuracy and locate losses with some degree of certainty. When, however, the number of blades on impellers is only four or five the mean path of the water through them becomes very uncertain, and thus the problem of locating losses proportionately harder. The problem depends on many conditions, even in pumps of the same form and size, the head pumped against, the quantity pumped, and the velocity of rotation of impellers, all of which are interdependent and may vary widely.

Here, then, is a problem in which theory can only suggest probabilities and experiment must decide.

In each of the main pumps the writer has attempted to follow the path of the water as best he could from the suction pipe into and along runner at the periphery and through vortex chambers to discharge pipe. The results are not satisfactory for the reasons already stated; however, they suggested a few points which will be mentioned.

Radial flow to impeller was assumed; its mean value was obtained from the dimensions of pump and knowing the quantity of water pumped. This velocity, together with the known velocity of rotation at inner ends of runners, should determine the angle at which the water enters the blades and show the amount of shock, if any, provided that the assumption of radial flow is correct.

As a matter of fact we know that this assumption is not absolutely correct, as shown by the wear at the intake of the pump casing.

The amount of this whirl that is set up before water enters blades of impellers was not measured and is not known. It is probably less on those pumps provided with plates in suction elbows to prevent such whirl. However, this assumption of radial flow offers a working hypothesis.

On this basis the *Delta* shows by far the greatest loss in shock at intake, while the *Gamma* shows the least, the *Epsilon* being next to the *Gamma*. The runner of the *Gamma* has only 4 blades, which makes the results possibly less reliable than in the *Epsilon* and *Delta* with 5 blades.

Any sudden change of velocities in the water along its path through the pump is accompanied by losses due to shock and to eddies. The velocities throughout the pump of the *Epsilon* are far more uniform than for any of the other cases. Among the other pumps, possibly the *Gamma* has more uniform velocities throughout the pump than any other. However, there are sudden changes in all.

It is known to be a fact that where a small number of blades are used on impellers cobblestones have been retained for some time behind runners instead of being thrown out radially. This, of course, could only be due to eddies which

would be largely overcome by increasing the number of blades, but the increase of surface would cause greater surface-friction losses.

In another class of hydraulic machines, viz, turbines, it has been found desirable to use a great many blades. The question of the proper number of blades for a centrifugal pump can best be settled by experiment. The writer does not know of any such experiments having been made.

The efficiencies obtained in every case, except that of the *Delta*, are very satisfactory and fairly uniform. Undoubtedly they are not absolute efficiencies, but it is believed that they are as close to the truth as it is possible to arrive by careful experimentation.

It is a noteworthy fact that the *Gamma* gave almost identical results for the efficiency tests of 1897, made while pumping sand, and the tests of 1902, made while pumping water, when efficiencies are corrected in the former.

The pump was the same in both tests, having been merely kept in repair since the first tests. However, the "total head" for the two sets of tests was quite different.

The *Epsilon* and *Zeta* showed much higher efficiencies in 1897, when they had impellers with 7 blades and the angles of intake of these blades different from that on the 5-blade impellers now in use. The total head and the revolutions per minute all vary, and it is therefore impossible to draw any definite conclusions from a comparison of the results of these two sets of tests. However, the great difference between the revolutions of the *Delta* may account for the low efficiency of the pump in the test of 1902, at least partially. In both cases the *Delta* shows the lowest efficiency.

Highest average efficiency was attained with a pump having a vortex chamber circular in cross section, while the lowest was attained by a pump having a rectangular cross section for its vortex chamber. The highest individual efficiency was attained on the *Flad*. Both the *Kappa* and the *Flad*, with their square casings, showed good average results.

The writer has failed to discover any obstacle worth considering to the use of a double suction. Certainly from the mechanical standpoint the perfect balance resulting makes it highly desirable.

The single-suction pump showed the lowest efficiency. This, however, does not prove anything, as proportions of casings, shapes of impellers, total head, etc., varied widely in the different pumps.

JET PUMPS.

A comparison between the compound-condensing, direct-acting steam pumps and the centrifugal pumps driven by means of compound-condensing engines should include the relative consumption of steam in the two types of engines. The former, in the experience of the writer, requires about 33 pounds of steam per horsepower hour, while the latter requires between 22 and 25 pounds.

The probable gain in steam consumption by using the high-speed compound-condensing engine, in average cases, will not be far from 25 or 30 per cent.

The greater steam economy of the engine of the centrifugal pump will thus partially offset the superior efficiency of the reciprocating pump. The slip and leakage in the reciprocating pumps may also be quite an appreciable quantity.

LOSS OF HEAD IN DISCHARGE PIPE.

The loss of head due to the double elbow at the stern of dredges *Kappa*, *Iota*, and *Flad* varies somewhat. This point has been discussed by Mr. Maltby.

The discharge pipes have countersunk rivets and are polished smooth and bright by the sand. The only disturbing influences in the flow of the water with a straight discharge pipe are due to the lapping of the various sections where they are riveted and to the flexible rubber joints connecting the different lengths of pipe.

These pipes are made up of sections, each being inserted into the succeeding section, thus making the mean area of pipe at the center of the section, where all these piezometer pressures were measured.

In case discharge pipes are not straight, there is loss due to the curvature of these flexible joints. The drop along the discharge pipe of the *Flad* shows the latter loss to be small even in the extreme case.

Mr. Maltby has given the data in the case of the four air cocks placed on the "quarters" of a section of pipe. It is to be remembered that these air cocks were purposely placed to show the greatest possible error, and that it is extremely

improbable that any such variation as they showed ever occurred in the data obtained in the tests.

As there is very little data on losses in pipes of this size, and at high velocities obtained, the results are of peculiar interest and value.

Using the formula $H_f = \frac{4fL}{D}\frac{v^2}{2g}$ or $f = \frac{DH_f}{4LHv}$, we may compute from the data given the value of the coefficient of friction f, for the pipes, including the flexible rubber connections.

The values of f as again given in the above formula, and obtained in most cases from mean values, are shown in the table below:

Name of dredge.	Diameter of discharge pipe.	f.
	Inches.	
Gamma	34	0.00445
Delta	$34\frac{1}{2}$	.00547
Zeta	32	.00569
Iota	$32\frac{11}{16}$	.00455
Kappa	$31\frac{7}{8}$	.00412
Flad	32	.00556
Average		.00503

The fact that zero pressure was found in the side of the discharge pipes of the *Iota*, *Kappa*, and *Flad* is accounted for by the fact that the water was flowing at a constant velocity in an open pipe, the only force retarding it being the skin friction and viscosity of water. The pressure will be zero when we go back from the discharge end of pipe to a section at which the force necessary to overcome this friction was just balanced by the head due to the open end of the pipe. The piezometer was at the center of the pipe, so the pressure ought to be neutral at such a distance as will make $H_f = \frac{D}{2}$. Results show this distance to be approximately 30 feet.

AMOUNT OF SAND IN MIXTURE PUMPED.

It is interesting to find the sand so thoroughly mixed with the water; the results indicate that there was slightly more on the lower side than on the top side of pipe, but that the mixture was fairly uniform.

If the pipe were inserted in discharge pipe at such a point that the static pressure were great enough to give the water in sampling pipe the same velocity as the water surrounding this pipe has on its way to the end of discharge pipe—in other words, at such a point that it is equally easy for water to escape from sampling and discharge pipes—there could be no doubt about obtaining an average sample.

Respectfully submitted.

W. B. Gregory, *Consulting Engineer.*

Capt. Wm. B. Ladue,
Corps of Engineers, U. S. Army,
Secretary Mississippi River Commission.

ADDITIONAL REPORTS OF ASST. ENGINEER F. B. MALTBY.

MEMPHIS, TENN., *September 18, 1903.*

CAPTAIN:

I have the honor to submit the following report of the results of tests made of the efficiency of the main centrifugal pump of the dredge *Zeta* with runners having different numbers of blades. This report to be supplemental to my report of May 7, 1903.

The tests described were made in obedience to the instructions from the Committee on Dredges contained in your letter of May 8, 1903, as follows:

"1. Install old 7-bladed runner on the *Zeta* and make tests with the same under the conditions when the runner now in use (5-bladed) was tested.

"2. Alter the pattern of the *Epsilon's* and *Zeta's* runner, if practicable, so as to cast a runner with three blades, and test this in the same manner."

It was found impracticable to comply with the instructions contained in the second paragraph in so far as altering the old pattern and a new pattern was made, from which a runner having 3 blades was cast.

A *drawing is submitted herewith, showing the outline and dimensions of the pump casing and of the runners with 3, 5 and 7 blades, and attention is invited to the following characteristics:

The pump has a double suction and discharges horizontally at the bottom; it has a vortex chamber of a spiral form in elevation and with an approximate elliptical section. The sides of the casing between the vortex chamber and the short curve leading to the suction openings are straight.

The runners all have an outside diameter of 69 inches at the tips and a diameter of 30½ inches at the inside shoulder of the blades. The degree of curvature of the face of the blades is the same in each case. The angle which the face of the blades makes with a tangent to the circle at the inside shoulder of the blades is the same on the 3 and 7-bladed runners, but is somewhat sharper than on the one having 5 blades. It will be noticed that the inside shoulder of the blades is considerably nearer the shaft than the straight portion of the casing, owing to the worn condition of the curved throat casting, but whatever effect this may have on the efficiency of the pump would be relatively the same with each runner.

The amount of clearance between the edges of the blades and the casing was adjusted in each case by the use of edge plates, fitted to the shape of the casing and bolted to the face of the blades. On the runner with 5 blades, which is the one in use pumping sand at present, these edge plates are of cast iron and have a thickness on the edge of about two inches. These edge plates give the face of the blade as a whole a trough shaped section, as shown. The edge plates used on the 3 and 7-bladed runner were plain steel plates, only about ¼ inch in thickness, and the face of the runner is flat. The clearance in each case was from $\frac{1}{16}$ inch to ⅛ inch.

METHODS.

The instruments used and methods employed in making observations were the same as described in my previous report as referred to above. Conditions of operation were as nearly the same during the tests of each runner as it was practicable to make them. One thousand feet of discharge pipe was used and the same instruments were applied in the same positions in each instance.

* Not printed; see Plate 38 for details of pump and 5-bladed runner.

RESULTS.

A table of data and results is submitted herewith. Each result is the mean of about twenty observations.

In the mean results for each runner it will be observed that certain figures have been omitted. This was done, not because of any lack of confidence in the figures, but because they covered speeds widely different from those observed with the other runners.

The mean efficiency of engine and pump with 3 blades is 71.6 per cent; with 5 blades, 72.1 per cent; and with 7 blades, 73.1 per cent; a total difference of only 1½ per cent. The amount of work done in each case varies considerably, being 231,475 foot-pounds per second with 3 blades, 282,589 foot-pounds per sec-second with 5 blades, and 288,518 foot-pounds per second with 7 blades; or the 5-bladed runner did 22 per cent more work than the one with 3 blades. The one having 7 blades did 24 per cent more work than the one having 3 blades, but only 2 per cent more than the runner with 5 blades.

CONCLUSIONS.

From the above results it appears that, considering mechanical efficiency alone, it makes very little difference, within reasonable limits, how many blades there are on a runner, though the greater number has a slightly higher mean.

The capacity of the plant, however, is very much more with the higher number of blades, though the difference between 5 and 7 is small.

Apparently the plant ran more smoothly and with less vibration with the 7-bladed runner than with the 3 or 5-bladed one, and the gages fluctuated very much less, showing that the work being performed, or the output of the pump, was more uniform.

Very respectfully,
Your obedient servant,

F B. Maltby,
Assistant Engineer.

Capt. Wm. B. Ladue,
Corps of Engineers, U. S. A.,
St. Louis, Mo.

Data and results of main pump tests of Dredge Zeta.

Revolutions per minute.	Steam pressure Lbs.	Total I. H. P.	Suction pipes. Mean velocity. Ft. per sec.	Suction pipes. Velocity head. Ft. of water.	Discharge Pipe. Center velocity. Ft. per sec.	Discharge Pipe. Mean velocity. Ft. per sec.	Discharge Pipe. Velocity head. Ft. of water.	Discharge. Cu. ft. per sec.	Discharge. Lbs. per sec.	Heads. Ft. of water. Suction.	Heads. Ft. of water. Delivery.	Heads. Ft. of water. Velocity.	Heads. Ft. of water. Total.	Work done. Ft. lbs. per sec.	Efficiency of pumps and engine; per cent.	Efficiency of pump; per cent.	Peripheral velocity. Ft. per sec.	
178....	120	592.08	13.701	2.92	19.397	15.735	3.85	87.880	5492.50	8.22	33.86	0.93	43.01	236232.43	72.5	78.8	53.590	3-bladed runner.
180....	130	622.08	13.727	2.93	19.434	15.765	3.86	88.048	5503.00	7.99	34.65	0.93	43.57	239765.71	70.1	76.3	54.192	
178 ...	121	591.51	14.644	3.33	20.732	16.818	4.40	93.929	5870.56	7.85	33.83	1.07	42.75	250966.44	77.1	83.8	53.590	
180....	130	640.94	13.819	2.97	19.565	15.871	3.92	88.640	5540.00	7.58	35.02	0.95	43.55	241267.00	68.4	74.3	54.192	
176 ...	137	551.12	13.164	2.69	18.636	15.118	3.55	84.434	5277.13	7.24	32.14	0.86	40.24	214351.71	70.0	76.1	53.068	
169....	127	524.47	13.026	2.64	18.442	14.960	3.48	83.552	5222.00	7.08	31.58	0.84	39.50	206269.00	71.5	77.7	50.880	
Mean..	......	587.03	13.680	2.91	19.334	15.711	3.84	87.746	5484.20	7.66	33.51	0.93	42.10	231475.38	71.6	77.8	53.253	....
*150....	132	483.85	12.321	2.36	17.443	14.150	3.08	79.028	4939.25	5.85	29.81	0.72	36.38	179689.91	67.5	72.9	45.1605	5-bladed runner.
*159....	134	571.15	13.067	2.65	18.500	15.007	3.50	83.814	5238.88	6.47	33.75	0.85	41.07	215140.26	68.4	73.9	47.8700	
170 ...	126	690.40	14.234	3.15	20.150	16.347	4.15	91.298	5706.12	7.72	39.60	1.00	48.32	275719.72	72.6	78.4	51.1819	
171....	125	712.06	14.304	3.18	20.250	16.428	4.19	91.750	5734.37	7.85	41.29	1.01	50.15	287578.65	73.4	79.3	51.4830	
175 ...	129	734.07	14.347	3.20	20.310	16.477	4.22	92.024	5751.50	8.28	40.16	1.02	49.46	284469.19	70.4	76.0	52.6873	
Mean..	....	712.18	14.295	3.18	20.237	16.417	4.19	91.691	5730.66	7.95	40.35	1.01	49.31	282589.19	72.1	77.9	51.7831	......
180....	135	764.52	15.235	3.61	21.603	17.496	4.76	97.715	6107.19	6.90	40.73	1.15	48.78	297908.1	70.85	77.0	54.192	7-bladed runner.
180....	135	770.15	15.643	3.80	22.182	17.965	5.02	100.334	6270.87	7.13	41.83	1.22	50.18	314672.4	74.28	80.7	54.192	
168....	125	639.47	14.769	3.39	20.942	16.961	4.47	94.727	5920.44	6.13	37.35	1.08	44.56	263814.2	75.01	81.5	50.759	
170....	133	665.83	14.854	3.43	21.063	17.059	4.52	95.275	5951.69	6.28	38.22	1.09	45.59	271474.3	74.14	80.6	51.1819	
*161....	132	569.33	14.017	3.05	19.875	16.097	4.03	89.902	5618.88	5.73	34.46	0.98	41.17	231329.3	73.87	80.3	47.4115	
180....	137	754.83	15.294	3.64	21.687	17.564	4.80	98.095	6130.94	6.86	41.03	1.16	49.05	300721.2	70.79	76.9	54.192	
175 ...	128	703.53	15.093	3.54	21.402	17.333	4.67	96.805	6050.31	6.56	39.17	1.13	46.86	283517.5	73.27	79.6	52.566	
Mean..	......	716.39	15.148	3.56	21.480	17.396	4.71	97.158	6072.41	6.64	39.72	1.14	47.50	288517.9	73.06	79.4	52.8462	....

* These lines omitted from mean.

NOTE.—In above tests with different runners all conditions were as near the same in each case as possible. Diameter runner 69″ at tip of blades—2 suction pipes $24\frac{1}{4}$″ inside diam. Discharge pipe 32″ inside diam., 1000 ft. long. Efficiency of pumps alone based on 92% efficiency of engine.

MEMPHIS, TENN., *October 30, 1903.*

CAPTAIN:

I have the honor to enclose herewith a copy of the results of the boiler test of the dredge *Beta*, made October 23, 1903. It will be noted, in comparison with the tests submitted with my report on tests of dredges, that the efficiency is slightly less than on the *Delta*. The difference is very small and I think, therefore, that the tests of the same boilers on the two dredges represent very well the actual efficiencies of this class of boilers.

I do not see how a better test than we got on the *Beta* could be devised. Everything moved very smoothly and uniformly during the whole duration of the test. The amount of water pumped per minute varied hardly any for hours at a time. The coal went in with almost equal regularity.

Very respectfully,

Your obedient servant,

F. B. MALTBY,

Assistant Engineer.

CAPT. WM. B. LADUE,

Corps of Engineers, U. S. A.,

St. Louis, Mo.

Results of boiler tests.

Dredge *Beta*.

Date of trial, October 23, 1903. Duration of trial, 9 hours.

Type of boiler, 2-Heine water tube.

Grate surface, 109 square feet. Water heating surface, 5,436 square feet.

Proportion of grate to heating surface, 1—50.

Proportion of air space in grate,

Average pressures, temperatures, etc.	
Steam pressure in boiler by corrected gage, 173.3	165.3 lbs.
Temperature of feed water entering boiler	166.1 degrees.
Temperature of gases escaping from boiler	637 degrees.
Draft gage, inches of water	$\frac{8}{10}$
Percentage of moisture in steam	21%
Fuel.	
Total amount of coal consumed	28,066 lbs.
Total amount of refuse, dry 2,645 lbs	9.4 %.
Total amount of combustible	25,421 lbs.
Coal consumed per hour per square foot grate surface	.28$\frac{6}{10}$ lbs.
Coal consumed per hour per square foot heating surface	
Water.	
Total water pumped into boiler and apparently evaporated, corrected for difference of level in boiler, leakage and steam pressure	216,163.5 lbs.
Water actually evaporated, corrected for quality of steam	211,523 lbs.
Equivalent water evaporated into dry steam from and at 212 degrees	232,464 lbs.
Equivalent water evaporated into dry steam from and at 212 degrees, per hour	25,829 lbs.
Equivalent water evaporated into dry steam from and at 212 degrees, per lb. of coal	8.28 lbs.
Equivalent water evaporated into dry steam from and at 212 degrees, per lb. of combustible	9.15 lbs.
Equivalent water evaporated into dry steam from and at 212 degrees, per square foot heating surface per hour	4.7 lbs.
Equivalent water evaporated into dry steam from and at 212 degrees, per square foot grate surface per hour	236.8 lbs.
Commercial horse-power developed, based on evaporation of 34½ lbs. of water from and at 212 degrees per hour	748 H. P.

MEMPHIS, TENN., *November 3, 1903.*

CAPTAIN:

I have the honor to enclose herewith a blue print of "Table Showing Results of Tests of Main Pumps".

This table has been revised to include the *Beta* and the *Iota* with the shrouded runner. It will be noted that the quantities for the *Iota* with the open runner have also been changed slightly from the original table, owing to an error in computation.

The *Beta's* efficiency at ordinary speeds of operation is only about 55 per cent, not quite as good as the *Delta.* I know of no reason for this except that it is due to the type of pump. The clearance between the runner and the casing is slightly more than on some of the other pumps, but I do not believe that it is enough more to account for all the differences in efficiency.

The table also shows the additional efficiency, and especially the capacity, obtained by putting the shrouded runner on the *Iota.* The efficiency of the engine and pump at 168 revolutions per minute is about 76 per cent, which is the highest efficiency we have yet obtained and is over 10 per cent greater than that obtained with the open runner. The average amount of work done is 341,500 foot-pounds per second, as against 310,800 with the open runner at the same speed, an increase of nearly 10 per cent.

In this connection I beg to invite your attention to the table, and especially in reference to the capacity of the pumps at different speeds. A diagram is enclosed, showing graphically the work done by each dredge in thousands of foot-pounds per second by each pump at varying peripheral velocities. In every case it will be noted that the capacity of the pump increases to a very marked degree to the maximum speed, with no indication that the speed for maximum capacity has been reached on any pump. The efficiencies at varying speeds are also plotted on the same sheet and show very little difference at different speeds. Generally speaking, the efficiencies are slightly lower at the higher speeds, except on the *Beta,* where the efficiency seems to increase slightly with the speed. This is true, also, on the *Iota.*

In the whole series of experiments, leaving out only the *Beta* and *Delta,* whose efficiencies are so much lower than the others as to be very marked, the whole range of speed from 42 to 55 feet per second, or 30 per cent increase, has been followed by an extreme range of efficiency of from about 63 per cent to 75 per cent, while the range in capacities is over 200 per cent of the lowest. Evidently, then, regarding mechanical efficiency alone, speed, within reasonable limits, makes very little difference, but to obtain the maximum output from the plant the pumps should be run at as high a speed as is practicable.

Very respectfully,

Your obedient servant,

F. B. MALTBY,

Assistant Engineer.

CAPT. WM. B. LADUE,

Corps of Engineers, U. S. A.,

St. Louis, Mo.

Water tests.

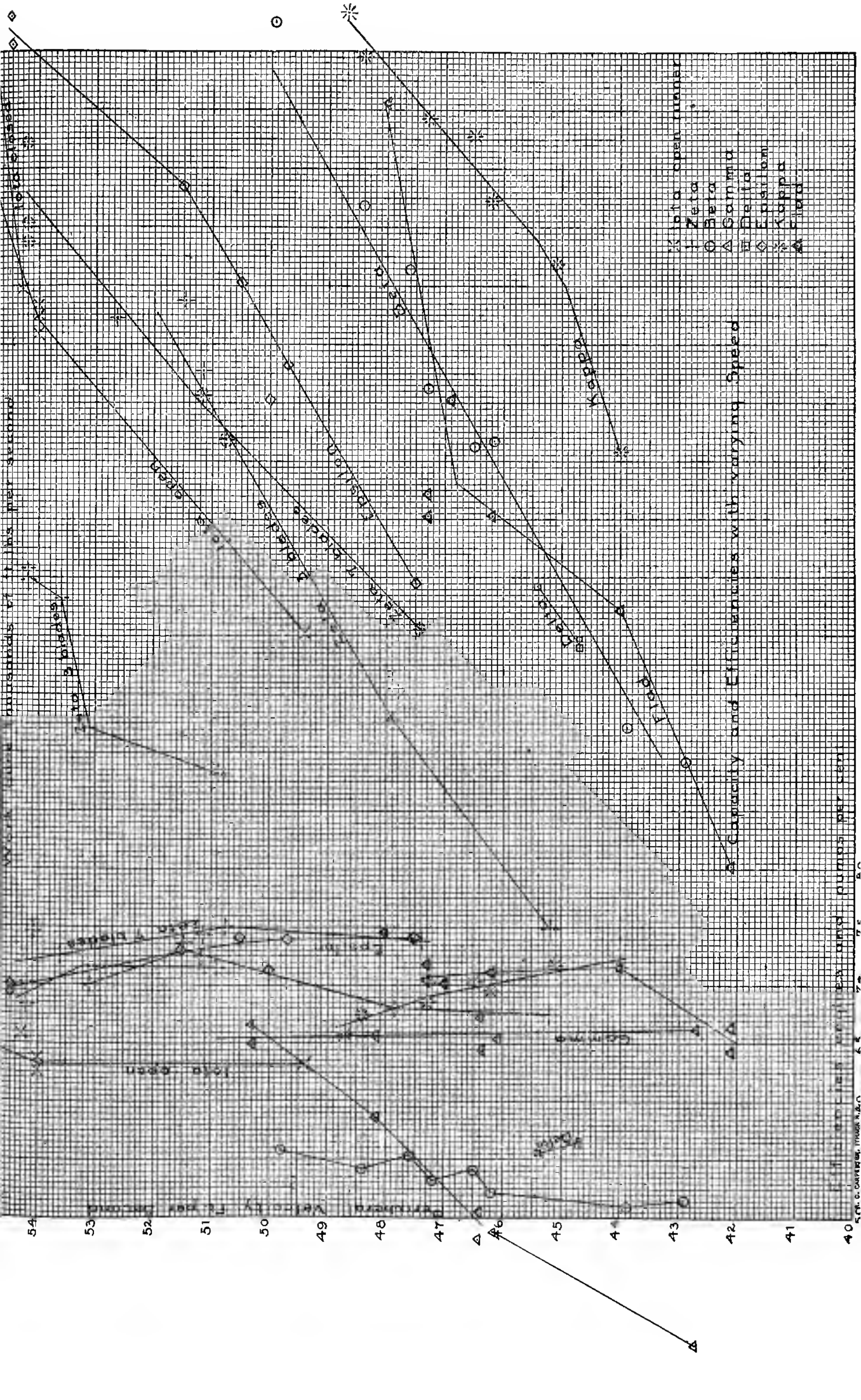
Data and results of main pump tests of Dredge Iota.

Revolutions per minute.	Steam pressure. Lbs.	Vacuum, inches mercury.	Total I. H. P.	Suction pipes. Mean velocity. Ft. per sec.	Suction pipes. Velocity head. Ft. of water	Discharge Pipe. Center velocity. Ft. per sec.	Discharge Pipe. Mean velocity. Ft. per sec.	Discharge Pipe. Velocity head. Ft. of water.	Discharge. Cu. ft. per sec.	Discharge. Lbs. per sec.	Heads, feet of water. Suction.	Heads, feet of water. Delivery.	Heads, feet of water. Velocity.	Heads, feet of water. Total.	Work done. Ft. lbs. per sec.	Efficiency of pumps and engine; per cent.	Efficiency of pump; per cent.	Peripheral velocity. Ft. per sec.	$\frac{\sqrt{2gh}}{V_p}$	
151	155	23	667.69	16.016	3.99	18.600	17.181	4.59	100.131	6258.19	9.74	26.55	.60	36.89	230864.63	62.9	72.4	49.4147	1.0181	500 ft. pontoons.
165	150	24.8	817.83	16.622	4.29	19.400	17.920	4.99	104.438	6527.38	10.71	31.75	.70	43.16	281721.82	62.6	72.2	53.9963	1.0086	
165	150	25	817.83	16.862	4.42	19.680	18.179	5.14	105.947	6621.69	10.68	31.85	.72	43.25	286388.08	63.7	73.3	53.9963	1.0090	Open runner.
165	150	25	817.83	16.751	4.36	19.550	18.059	5.07	105.248	6578.00	10.77	31.80	.71	43.28	284795.84	63.3	73.0	53.9963	1.0099	
168	146	25.2	867.54	17.393	4.70	20.300	18.751	5.47	109.281	6830.06	10.79	33.80	.77	45.36	309711.52	64.9	74.0	54.978	1.0092	
168	146	25.2	867.54	17.393	4.70	20.300	18.751	5.47	109.281	6830.06	11.22	33.70	.77	45.69	312065.44	65.4	74.6	54.978	1.0137	
166	147	24	799.10	16.75	4.35	19.37	18.10	5.01	105.52	6595.00	12.35	30.91	.66	43.92	289652.40	65.9	71.6	54.324	....	Shrouded runner.
168	155	24	823.49	19.85	6.12	22.65	21.30	7.05	124.18	7761.25	12.48	31.16	.93	44.57	345918.91	76.4	83.0	54.978	....	
168	153	24	800.82	19.45	5.86	22.20	20.90	6.77	121.85	7615.44	12.41	30.96	.91	44.28	337211.57	76.1	82.7	54.978	....	

Data and results of main pump tests of Dredge Beta. Water Test.

Revolutions per minute.	Steam pressure. Lbs.	Vacuum, inches mercury.	Total I. H. P.	Suction pipes. Mean velocity. Ft. per sec.	Suction pipes. Velocity head. Ft. of water	Discharge Pipe. Center velocity. Ft. per sec.	Discharge Pipe. Mean velocity. Ft. per sec.	Discharge Pipe. Velocity head. Ft. of water.	Discharge. Cu. ft. per sec.	Discharge. Lbs. per sec.	Heads, feet of water. Suction.	Heads, feet of water. Delivery.	Heads, feet of water. Velocity.	Heads, feet of water. Total.	Work done. Ft. lbs. per sec.	Efficiency of pumps and engine; per cent.	Efficiency of pump; per cent.	Peripheral velocity. Ft. per sec.	$\frac{\sqrt{2gh}}{V_p}$	
129	150	....	....	19.71	6.03	21.79	20.27	6.38	122.23	7639.38	10.72	24.58	.35	35.65	272343.90	....	....	47.2811	....	600 ft. pontoons
126	147.5	....	881.75	18.79	5.48	20.77	19.32	5.80	116.50	7281.25	10.59	23.89	.32	34.80	253387.50	52.2	56.7	46.1815	....	
127	147.5	....	882.19	19.35	5.81	21.39	19.90	6.15	120.00	7500.00	10.53	24.14	.34	35.01	262575.00	54.1	58.8	46.5477	....	
129	150	....	930.18	19.87	6.13	21.79	20.27	6.38	122.23	7639.38	10.62	24.76	.25	35.63	272191.11	53.2	57.8	47.2811	....	
132	145	....	1018.30	20.23	6.35	22.36	20.80	6.72	125.42	7838.75	11.36	27.01	.37	38.74	303673.18	54.2	58.8	48.3806	....	
130	142.5	....	962.97	19.87	6.13	21.97	20.43	6.48	123.19	7699.38	11.21	26.51	.35	38.07	293115.40	55.3	60.1	47.6477	....	
136	....	....	1102.47	20.91	6.79	23.12	21.50	7.18	129.65	8103.13	12.42	28.77	.39	41.58	336928.15	55.5	60.3	49.8467	....	
117	155	....	743.15	17.63	4.81	19.49	18.13	5.10	109.32	6832.50	9.59	20.66	.29	30.54	208664.55	51.5	55.9	42.8828	....	
120	152	....	777.74	18.23	5.16	20.16	18.75	5.46	113.06	7066.25	9.65	20.37	.30	30.32	214248.71	50.7	55.1	43.9824	....	

Dredge *Beta*.—Inside diameter of suction pipes 33¾ inches; area = 6.20 sq. ft. Inside diameter of discharge pipe 33¼ inches; area = 6.03 sq. ft. Diameter of pump runner 84 inches. The runner is open, with 8 blades of the Rankine type, see Plate 49.


Capacity and Efficiencies with varying Speed
Iota open runner
Zeta
Beta
Gamma
Delta
Epsilon
Kappa
Iota totally closed
Zeta 3 blades
Zeta 4 blades
Zeta 7 blades
Iota open
Kappa
Delta
Epsilon
Gamma
Peripheral Velocity Ft. per Second
54
53
52
51
50
49
48
47
46
45
44
43
42
41
40